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LONG-LIFE
ASSURANCE STUDIES
OF COMPONENTS

LONG-LIFE ASSURANCE STUDY FOR MANNED SPACECRAFT LONG-LIFE HARDWARE

Approved

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FOREWORD

This document is Volume III of a five-volume final report prepared by Martin Marietta Corporation, Denver Division for the National Aeronautics and Space Administration, Manned Spacecraft Center (NASA-MSC) under Contract NAS9-12359, *Long-Life Assurance Study for Manned Spacecraft Long-Life Hardware*. This study was performed with J. B. Fox, Manned Spacecraft Center, as Technical Monitor and R. W. Burrows, Martin Marietta, as Program Manager. Acknowledgment is made to the individual contributors identified in each volume and to R. A. Homan and J. C. DuBuisson, Task Leaders for the electrical/electronic and mechanical areas, respectively.

The five volumes submitted in compliance with Data Requirements List T-732, Line Item 4, are as follows:

Volume I - Summary of Long-Life Assurance Guidelines;

Volume II - Long-Life Assurance Studies of EEE Parts and Packaging;

Volume III - Long-Life Assurance Studies of Components;

Volume IV - Special Long-Life Assurance Studies;

Volume V - Long-Life Assurance Test and Study Recommendations.

Many of the issues discussed are controversial, and while the recommended guidelines are believed to represent the concensus opinion, it should be recognized that some guidelines may require tailoring to specific program constraints and objectives.

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I. INTRODUCTION

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A. STUDY OBJECTIVES

The objectives of this Long-Life Assurance Study were to develop and document the engineering approach necessary to assure that hardware selected for manned spacecraft is based on the experience gained from the manned and unmanned space programs conducted to date, and will meet the following long-life goal: A five-year operational lifetime goal without maintenance is considered the minimum long-life goal with a 10-year desired operational lifetime.

B. SCOPE

A detailed engineering study was conducted of the design; application; failure mechanisms; manufacturing processes and controls; screen and burn-in techniques; functional, qualification and life testing; and any other factors affecting the hardware items under study. The hardware items investigated are listed in Table 1. In addition, certain special studies were accomplished which are not oriented towards specific hardware items. The special studies are listed in Table 2 and presented in Volume IV.

C. APPROACH

The approach included a comprehensive review of the available technical data and an industry survey to establish a baseline for current hardware capability from which improvements for increased life and reliability goals can be assessed. Emphasis was placed on the review of failure history of the hardware as used in NASA and DOD manned and unmanned space and missile programs to ascertain the lifetime and corrective measures necessary to insure long-life operation.

Table 1 List of Hardware Items Studies

<u>EEE Parts and Packaging (Volume II)</u>

1) Monolithic Integrated Circuits
2) Hybrid Integrated Circuits
3) Transistors
4) Diodes
5) Capacitors
6) Relays
7) Switches and Circuit Breakers
8) Electronic Packaging

<u>Components (Volume III)</u>

9) Electric Motors and Bearings
10) Accelerometers
11) Gyroscopes and Bearings
12) Compressors and Pumps
13) Magnetic Tape Recorders
14) Plumbing Components and Tubing
15) Check Valves
16) Pressure Regulators and Solenoid Valves
17) Thermal Control Valves
18) Pressure Vessels and Positive Expulsion Devices
19) Ni-Cd Batteries
20) Transducers

Table 2 List of Special Studies

1) Temperature Cycling as Employed in the Production Acceptance Testing of Electronic Assemblies ("Black Boxes")

2) Accelerated Testing Techniques

3) Electronic Part Screening Techniques

4) Industry Survey of Electronic Part Derating Practice

5) Vibration Life Extension of Printed Circuit Board Assemblies

6) Tolerance Funnelling and Test Requirements

D. ORGANIZATION

This final report is divided into five volumes for convenience; viz:

1) Summary of Long-Life Assurance Guidelines;

2) Long-Life Assurance Studies of EEE Parts and Packaging;

3) Long-Life Assurance Studies of Components;

4) Special Long-Life Assurance Studies, and;

5) Long-Life Assurance Test and Study Recommendations.

The results of the specific component studies are presented in 12 separate chapters. The organization of each chapter is presented in Table 3. To prevent repetition, a detailed table of contents is not provided for each chapter. The exception is Chapter II on Electric Motors and Bearings which, because of its length, requires its own detailed table of contents.

Table 3 Volume III Organization

A. INTRODUCTION

B. GUIDELINES FOR LONG-LIFE ASSURANCE

   1. <u>Design Guidelines</u>
   2. <u>Process Control Guidelines</u>
   3. <u>Test Guidelines</u>
   4. <u>Application Guidelines</u>
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   1. <u>Failure Mechanism Analysis</u>
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      b. Results of Survey
      c. Alternate Approaches
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D. TEST METHODOLOGY AND REQUIREMENTS

   1. <u>Qualification</u>
   2. <u>Life Test</u>
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   4. <u>Burn-In</u>
   5. <u>Failure Mode Detection</u>

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   1. <u>Existing</u>
   2. <u>Needed</u>

F. PARTS LIST

   1. <u>Acceptable</u>
   2. <u>Unacceptable</u>

G. REFERENCES

II. ELECTRIC MOTORS AND BEARINGS

by S. Broadbent

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Figure

II. ELECTRIC MOTORS AND BEARINGS

A. INTRODUCTION

Reliability of a mechanism can be no better than that of its weakest link. In motorized mechanisms, that link is usually the motor for a variety of reasons, often because it is the fastest operating and most complex portion of the mechanism. Therefore, for assurance of long life, the motor must be given special consideration during the formulative stages of mechanism design. Concepts should evolve around the use of the optimum type of motor, rather than regarding the motor as an adjunct to be determined later.

A successful long life motor application is not simply a question of building a motor to the ultimate in quality standards; *it is equally dependent on utilizing the best selection of motor for the particular application and designing the driven mechanism appropriately*. This will involve numerous design iterations and analysis before optimization is accomplished. Situations will also be encountered where motor compromises will be necessary in order to enhance the driven mechanism. So, there is no universal panacea governing the selection of motors for space vehicle applications for the simple reasons that an immense variety of space environments exist; that duty requirements will vary from one short actuation per mission to continuous operation, that the type of function will vary, which, in itself, may dictate a limited species of motors; that the available power supply will have a powerful effect on the choice of motor type; and that space and weight constraints will also have important ramifications.

Each application must be given meticulous attention, both individually and in conjunction with other motor applications on the same vehicle. Special attention must be given to the choice of lubricants and bearings.

It must also be recognized that the word "motor" is an established term conventionally used for "motor assemblies" comprising not only a motor proper, but also integral accounterments such as gearheads, brakes, slip clutches, limit switches and hermetically sealed output drive mechanisms. More often than not, it is these mechanisms which are the life limiting and failure prone portions of the overall assembly.

Hence, with all the variables described above, it is not feasible to define a single set of selection criteria. Probably future space programs will be utilizing all of the many types and styles of motor and motor assembly currently available, but with state-of-the-art improvements. An intimate knowledge of prior experience is the first prerequisite in the determination of policies to improve upon that experience; or even to know if the prior art is in need of improvement. This applies not only to the art of motor design, but also to the art of motor application.

While motor design technology is largely locked in the minds of a few specialists and regarded as proprietary by the companies for whom they work, the prior art of motor application is available to any investigator adequately prepared to unfold it from the mass of documentation in which it is hidden.

It is submitted that a manual of motor application experience would be of invaluable assistance to the designers of motorized devices. To emphasize this need and to give perspective to this study, a review of current spacecraft motor applications is presented as part of this introduction. Special emphasis is given to applications and techniques conducive to long life. The illustrations selected for this review are also intended to facilitate the later discussion of "Fluid Lubrication Systems" and "Application Guidelines."

Historical Review of Spacecraft Motor Usage

This review will relate the needs of the various types of application and environment to the wide assortment of motors that have been used during the short history of this activity. Special attention will be given to those applications of significance to the achievement of long life in spacecraft.

It is based, in part, on a survey covering some 300 spacecraft motor applications. The survey is summarized in Martin Marietta Aerospace report T-71-48890-003. The percentage distribution of motor types is given in the following tabulation. These percentages apply to motor applications only. They do not take into account any repetition of the same application; i.e., the percentages do not define relative abundance of motor types. The 300 motor applications are believed to represent about 90% of the total unclassified spacecraft motor applications; hence, the percentages should be reasonably accurate. Thirty-five manufacturers shared in the supply of these motors. Of these, eight no longer cater to the space industry.

Type of Motor	Percent of Total
Hysteresis Synchronous	25.5
Permanent Magnet Stepper	18.2
PMDC & Wound Field Brush Type	16.5
DC Torquer Brush Type	12.0
Induction	11.0
Brushless Torquer PMDC	8.5
Variable Reluctance Stepper	4.0
Latching Stepper	2.9
DC Start AC Run	0.7
Brushless PMDC Motors	0.7

This may be restated:

AC	37.5
DC	37.4
Stepper	25.1

Approximately 34% of these motors were equipped with integral gearheads; and practically all of these were different; different design, manufacturer, gear ratio, power, and lubricant. While the competitive and specialized nature of U.S. industry is partially responsible for this variety, the basic reason (as previously mentioned) is the enormous diversity of application and environment.

Attempts to standardize and limit the number of motor types on a particular spacecraft have been only partially successful. Performance and weight penalties, plus interference in the freedom of choice on the part of subcontractors, have been the prime objections to standardization.

Hence, it is reasonable to assume that an equal, or even greater, diversification of motor types will be incurred in the future, but optimized for longer life capability.

The various motor types and their applications will be discussed in the order of the foreceeding list.

a. Hysteresis Synchronous Motors - A primary use for hysteresis synchronous motors has been in tape recorders to satisfy the requirement for precise speed control without the need of rate feedback. This choice also facilitated the achievement of a multi-speed capability, simply by changing the frequency of the applied voltage, meeting the requirements of a slow recording speed and a fast playback speed. In this application the motors are used without gearheads; OGO being an exception.

The majority of tape recorder applications have been satisfied with this type of motor usually of six or eight-pole construction, two-phase, 400 Hz supply. Of late, there appears to be a trend toward the use of a larger number of poles; 24 being an actual example. This reduces the number of belts used to produce the required capstan speed.

Almost without exception, the tape recorders have been hermetically sealed and subject only to limited temperature excursions (-10°C to +45°C). Under these conditions, hysteresis synchronous motors have given up to 26,000 hours operation before failure; failure being attributed to bearing lubrication exhaustion.

Both oil and grease lubrication have been utilized almost exclusively for tape recorders; temperature extremes have not warranted the use of dry film lubricant. Hysteresis synchronous motors have also been extensively used in scientific instruments where uniform speed is desirable; i.e., the activation of mirrors and choppers in spectrometers and radiometers. Motors with up to 80 poles have been used in these applications operating from two-phase, 400 Hz, square wave supply. These, also, did not involve gearheads.

Requirement for constant speed has been a primary reason for their selection in many instances, overruling any consideration of lower efficiencies. Their inherent simplicity, using a lightweight tubular rotor, is also very appealing. Furthermore, since there is no rotor slip (except during acceleration), there is no rotor heating as in the case of induction motors. For these reasons, the hysteresis synchronous motor has been selected for situations where a PMDC motor might appear to be a more logical choice, primarily to eliminate brush and EMI problems.

Modern multipole concepts can offer 60% efficiency combined with constant low speed operation. Their pancake configuration permits gear reduction to be incorporated integrally into the motor housing with minimal increase in envelope. Figure 1 is representative of this design philosophy. It will be observed that the motor rotor is bearingless.

The popularity and versatility of this type of motor is well illustrated in the Nimbus Program where 21 of 32 motor applications have been satisfied by hysteresis synchronous motors.

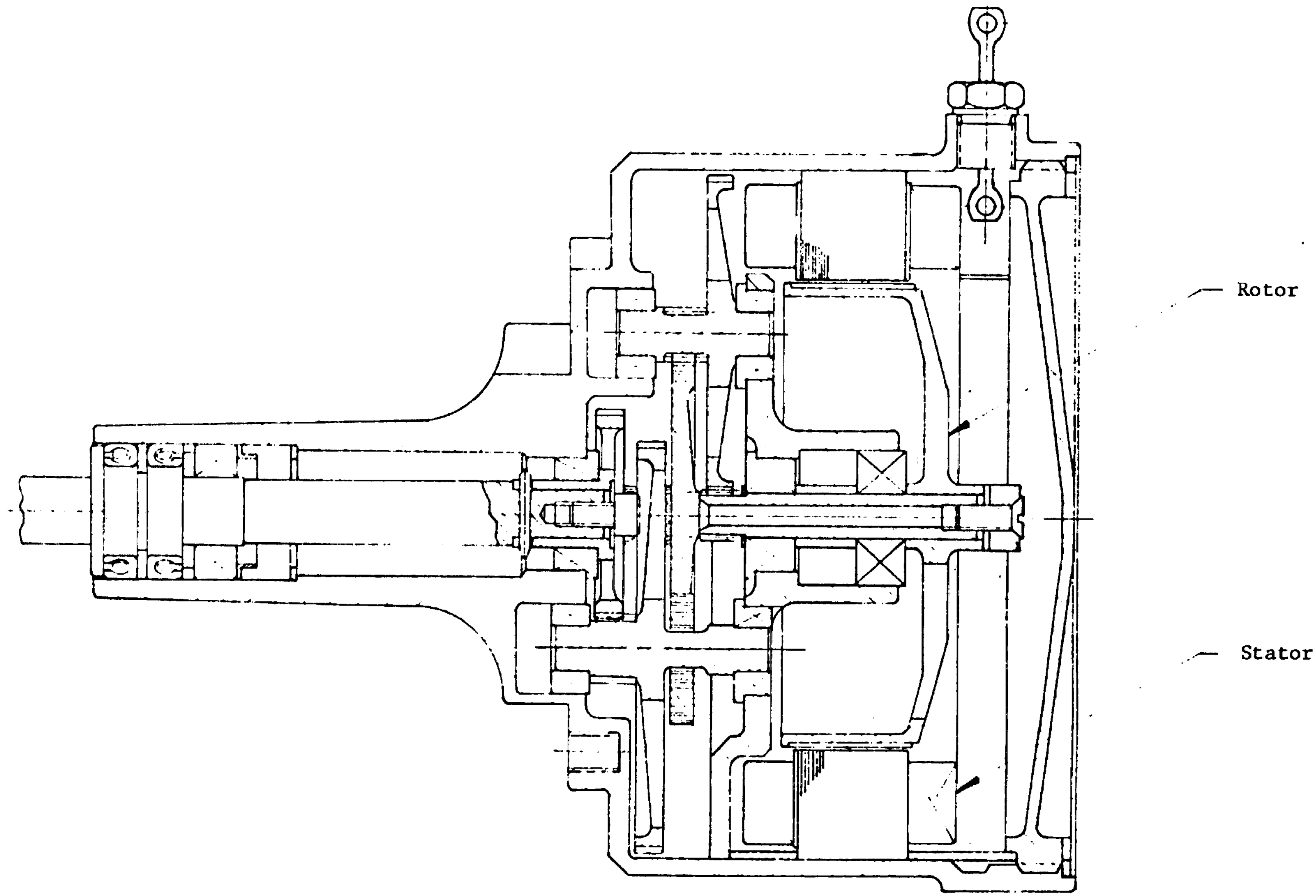

Figure 1 Multiple Hysteresis Synchronous Shaftless Motor with Integrated Gear Reduction

Courtesy of Schaeffer Magnetics, Chatfield, California

A particularly interesting application of a hysteresis synchronous motor is to the diffraction grating actuator for the Mariner UV Spectrometer. The motor operates at 9000 rpm and drives a planocentric hermetically sealed output stage via a 450:1 gear ratio. This is the only space usage of this form of hermetic shaft seal. The motor has Bartemp-solid transfer lubricated bearings, the gearing uses F50 Silicone oil. The unit has operated satisfactorily throughout its 90-day duty in orbit around Mars upon completing the 230-day trip from earth.

This concept is illustrated in Fig. 2. It can be used in conjunction with any type of motor. Somewhat similar concepts will be illustrated in connection with PMDC motors.

The life capability of the high speed ungeared type of hysteresis motors is illustrated by the "failure" of the record mode motor of the OAO II tape recorder after 26,000 hours of successful operation. This was located in the pressurized interior of the tape recorder and its bearings were wet lubed. Failure is attributed to loss of lubricant.

Though elementary, it is important to observe that the basic constructional features of the motor, i.e., the winding and rotor, did not degrade in any manner to cause this failure. Nor, in fact, were the bearings at fault. It was merely that an expendable medium, the lubricant, finally depleted to a point where it failed to provide an oil film of sufficient thickness to separate the surface asperities, resulting in increased heating, higher friction, oil degradation and varnish formation. Torque increase and higher motor currents are a manifestation of incipient bearing failure. This incipient failure mode may extend for as much as 10% of nominal lubricated life. It should not be relied upon even in applications where substantial increase in bearing torque is tolerable since the instant of final seizure following initial indication of failure is very unpredictable. Usually the rate of bearing degradation during this period is exponential.

Almost without exception, bearing/lubricant failure is the mechanism by which motors in space applications fail, regardless of type. The one exception is in the case of brush type motors where the brushes constitute a separate failure mechanism.

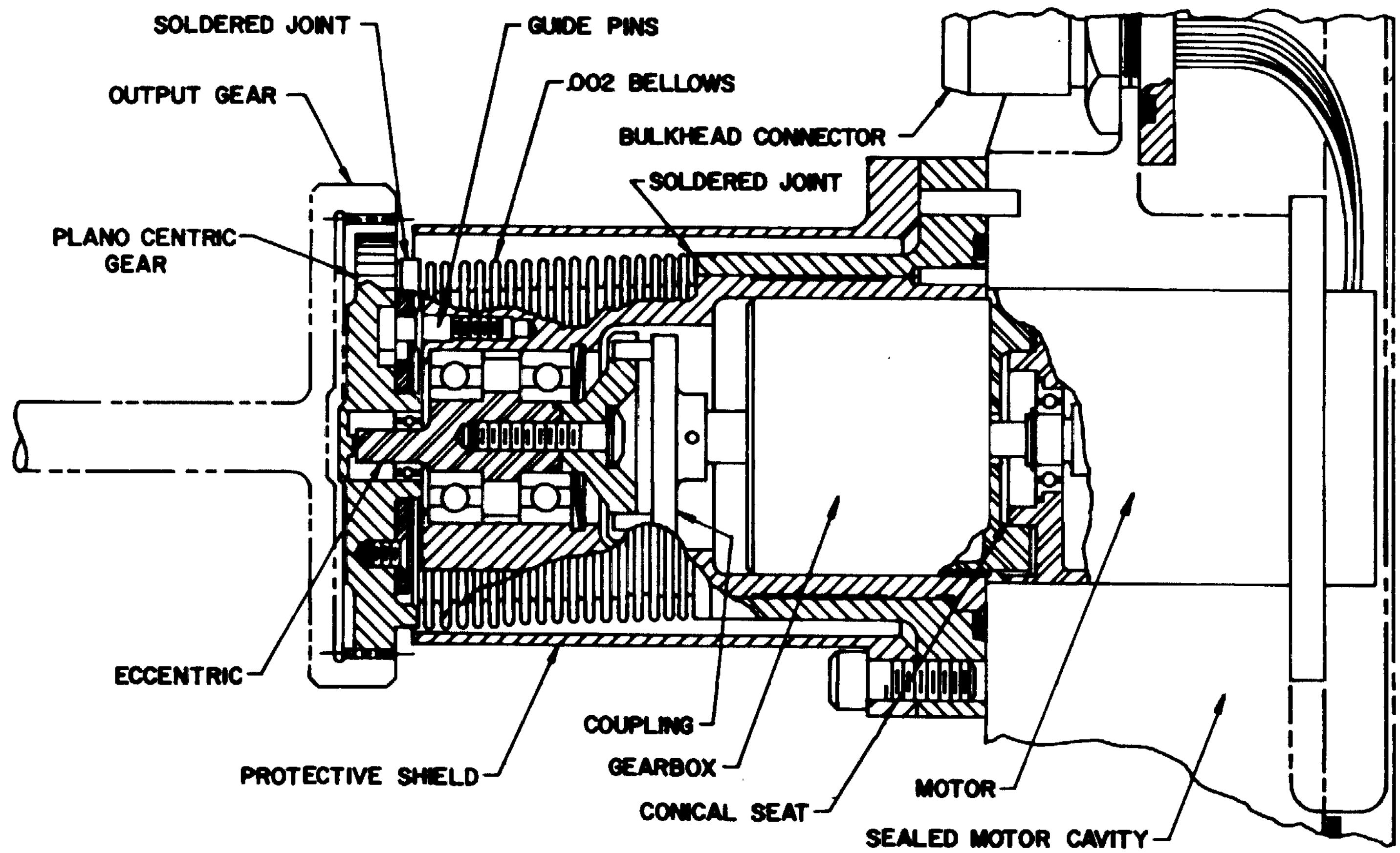

Figure 2 Hysteresis Synchronous Motor (Mariner UV Spectrometer. Courtesy of University of Colorado)

b. Permanent Magnet Stepper Motors - Permanent magnet stepper motors are the second most commonly used motor in space applications. They are usually no larger than size 15 ($1\frac{1}{2}$ in. diameter). As in the case of hysteresis synchronous motors, their primary value is their constant speed characteristics combined with simplicity.

A secondary characteristic, utilized in a number of applications, is the deenergized detent torque which, through the mechanical advantage of a gear ratio, can reach very useful proportions.

A substantial percentage of these motors have been either equipped with gearheads or operate through the medium of gear trains. Also, many have embodied Bartemp type bearings involving solid transfer lubrication in the form of Duroid retainers.

Stepper motors compare poorly with other types on a power to weight basis. It is only when their stepping or constant pulse rate (speed) characteristics are needed that they prove beneficial; except at power levels below that of the smallest practical size of conventional motor.

Their primary use has been in optical and other scientific instruments and shutter mechanisms in cameras. Usually, these motors have been of the conventional 90° step angle. More often than not, they have been adaptions from standard catalog items.

Antenna despin motors for ATS III and Intelsat IV are important examples of specially designed stepper motors. With the encoder embodied on the same shaft, they are equivalent to and the precurser of the brushless DC motors (using resolvers for commutation) now in common use for antenna despin function, e.g., Telsat, Skynet and Helios. These motors are of pancake construction with a three segment 128 pole stator and bifilar wound for redundancy. The despin problem on ATS III was due to thermally induced failures in the motor electronics, not to any shortcoming of motor or lubrication. This motor concept is subject of a NASA Goddard patent.

Another stepper motor application of advanced design is the Biaxial Antenna Drive on DSCS 2. In this case, the motor is a conventional size 15, 90° step angle unit. It operates a harmonic drive through a 23.6:1 gearhead. The unit is not hermetically sealed; labyrinth seals are used to retain the Bray Oil NPT4 lubricant, which is used for motor, gearing and harmonic drive.

A particular feature to beware of when using PM Stepper Motors over a broad temperature range, is demagnetization of the rotor. This can occur if it is stalled within a magnetic field greater than that at which it was stabilized. This can be prevented by using a constant current mode of energization, avoiding excess current draw at low temperatures which is the main danger of demagnetization. Alternatively, use high energy magnetic materials such as Samarium Cobalt or Alnico 9 which resist such demagnetization.

c. PMDC and Wound Field Brush Type Motors (Excluding Torquers) - The majority of the high speed brush type motors used on spacecraft have been of the permanent magnet field (PMDC) variety. Notable exceptions have been the Surveyor Soil Sampler Motors and the Wheel Drive and Steering Motors of the Lunar Rover.

The virtue of wound field motors is that by using a double field winding, reversing can be accomplished with a single-pole switch using three wires. PMDC require double throw switches but only two wires. Also, on low response servo-systems they simplify amplifier design.

The major attribute of this type of motor (either wound field or PM) is that its power to weigh characteristics are almost invariably superior to any other type. It also exhibits a greater reserve of stall torque. Another advantage is that it will operate directly from the customary dc spacecraft power source; whereas, other types may require inverters or pulse generators.

The need for brushes is the main objection to the use of this type of motor, particularly in vacuum or inert gas since both these conditions accelerate brush wear enormously. An associated shortcoming is the need for EMI suppression of brush noise.

High altitude graphite brushes, with various adjutants, and silver graphite compacts sometimes with molydenum disulphide, provide a life expectancy varying from 50 to 200 hours* in vacuum. While this is more than adequate to meet the needs of single-shot functions, such as the deployment of antennae, solar panels or stabilization booms, and many other short term operations, it does eliminate the use of unsealed high speed dc brush type motors for extended service. For instance, the maximum time registered on any

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*Dependent on commutator speed, number of bars and current density. One Manufacturer, General Design, Inc., using a proprietory brush material which is 70% copper can achieve 400 hour life at high speed and high current density--as proven on classified applications.

of Surveyor Soil Sampler Motors was 22.8 minutes; Boeing Type 046-045 compact brushes were used which are mostly MoS_2--excellent in vacuum, but poor for atmospheric conditions. Since motors are usually operated to a greater extent during ground testing than they are in orbit, brushes must cater to both earth and space environments unless pressure sealed.

Also, in the majority of applications for which these motors have been selected, substantial speed variations have been of little concern. This is important since the speed of dc motors is particularly sensitive to temperature and voltage changes--more so than other types.

d. DC/AC Design Mutation - A very interesting adaptation of a brush type dc motor was for tape recorders on OSO 1, Tiros and UK 1. This used brushes for starting on regular dc, then switched to 100 Hz pulsed dc as the brushes were lifted from the commutator by a centrifugal mechanism. It then functioned as an induction motor. This arrangement provided the high starting torque of a dc motor (without the brush wear problem), plus the advantage of close speed control when functioning as an induction motor.

On OSO 2, the same type motor operated the Scanning Disc on the White Light Coronagraph and was hermetically sealed using a cylindrical type permanent magnet coupling.

It is not known why this principle, which enjoyed the "best of both worlds" did not get greater use; the pulsed dc would need minimal electronics, and EMI suppression would be unnecessary for the very short periods that the brushes are operative. The supplier of this motor is Brailsford.

e. Hermetic Varients - There are several advantages of hermetic sealing a motor:

1) The evaporation rate of the lubricant is substantially retarded as a result of maintaining a pressurized atmosphere. Oil film and oil reservoir content will, therefore, remain intact for longer periods.

2) Outgassing (primarily from the lubricant) no longer constitutes a hazard to lenses and optics. There is no loss of lubricant.

3) The motor is isolated from contamination and corrosion hazards associated with moist and dusty environments during prelaunch and post landing operations.

4) The possibility of cold welding of rubbing surfaces (gears and bearings) is minimized.

5) Organic contamination of biological instrumentation is avoided.

When applied to dc motors, it must be realized that hermetic sealing will enhance brush life only when it is feasible to utilize a moisture content in the encapsulated gas, water vapor being essential to a satisfactory sliding interface between graphite and commutator. By this means, brush life can be extended up to 1000 hours. The shortcoming is that the operating temperature limit must be held above the dew point of the water vapor. Since an absolute humidity of two grains of H_2O/lb of air is required, and assuming pressurization to one-half atmosphere, the low temperature limit will be in the region of -30°F. Hence, there will be many motor applications unable to use this principle.

Two manufacturers have developed hermetically sealed dc motors which have seen service in space. Unfortunately, the duty requirement in both instances were so short that in all probability a non-hermetically sealed motor would have sufficed. These two motor concepts are illustrated in Fig. 3 and 4.

The concept of Fig. 3 used an OAO II for operation of the sunshade for the telescope, utilizes a wobble bellows to provide 90° to 95° angular travel. Limit switches, mechanical stops and a slip clutch are incorporated. This is the only space application of this motor.

The other arrangement (used on RAE I for boom deployment) utilizes a bellows in similar fashion to that illustrated in Fig. 4 except there is no final planocentric orbiting gear arrangement. The crank drives the output shaft directly. It will be noted that the hermetic sealing protects the high speed portions of the design, but leaves the output shaft bearings open to the environment. Hence, if the hermetic unit is intended to eliminate outgassing, the output bearings must be dry film lubricated. The ball detent slip clutch (torque limiter) is an optional feature. This motor was actuated after 2½ years of dormancy in orbit and functioned identically to its original performance, but then so did an unhermetically sealed PMDC motor after 15 months in orbit on a classified satellite.

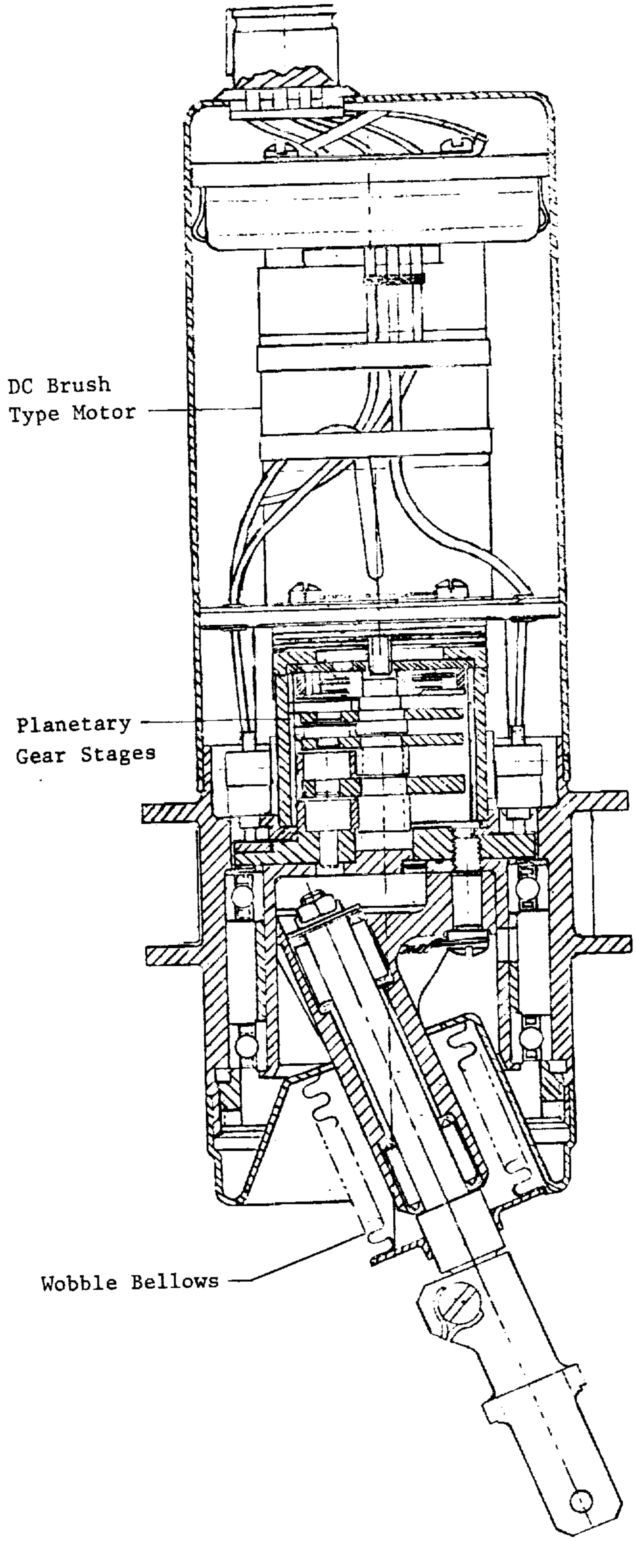


Figure 3 *Hermetically Sealed (DC) Motor using Wobble Bellows with Partial Rotation*

Courtesy of Lundy Electronics & Systems Glen Head, New York

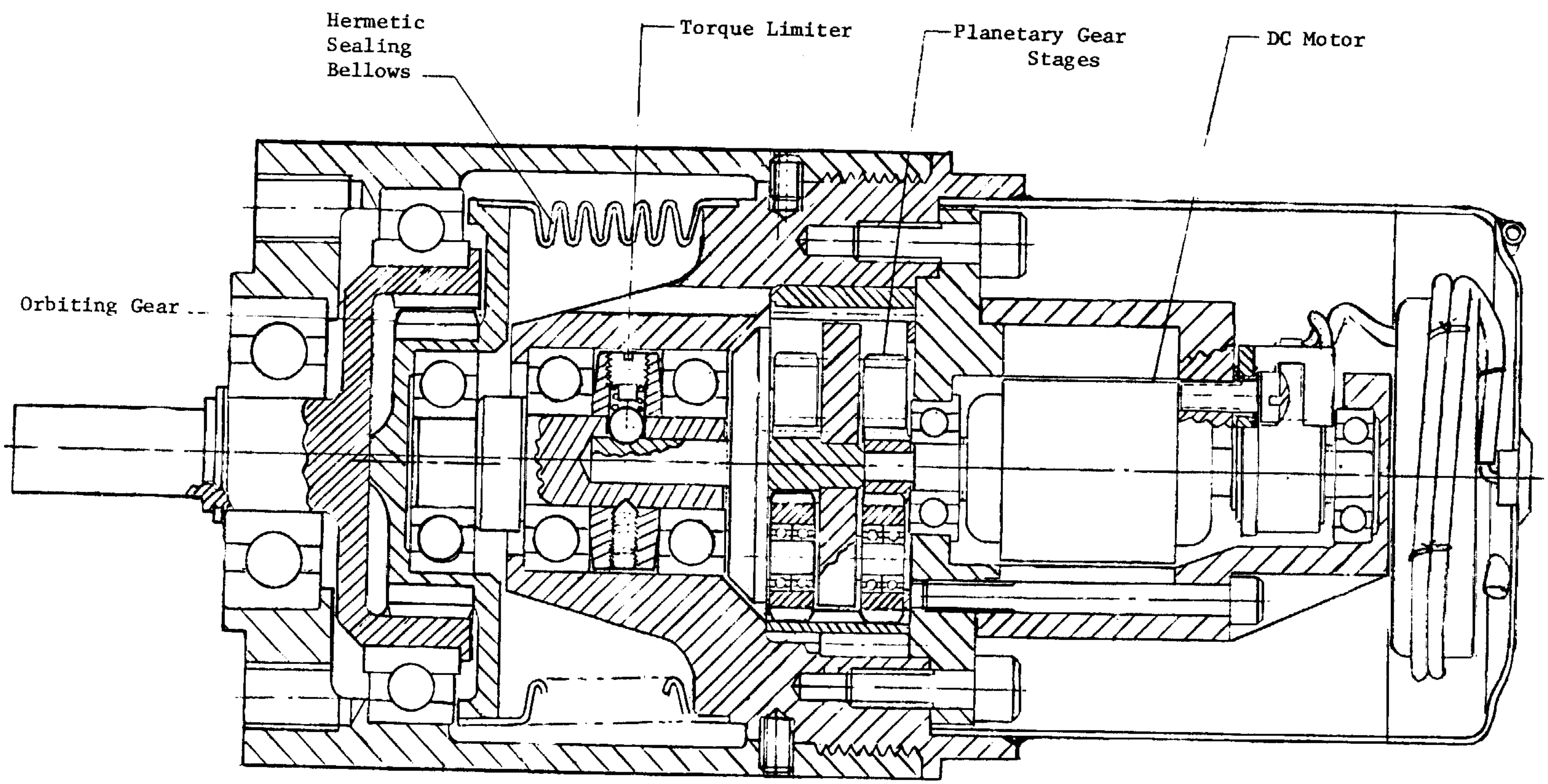


Figure 4 Hermetically Sealed (DC) Motor Using Planocentric Output Stage with Orbiting Gear

Courtesy of Nash Controls Div Simmonds Precision

Two other hemetically sealed units are in course of development. These are illustrated in Fig. 4 and 5.

The concept of Fig. 4 is basically equivalent to that of Fig. 2, except that the bellows is used to restrain the orbiting gear from rotating. Whereas, in the concept of Fig. 2, the orbiting gear is provided with radial grooves which engage stationary pins, thus, alleviating the bellows of a torsional load. Again, the output shaft bearings are open to space vacuum.

The arrangement of Fig. 5 utilizes a magnetic coupling between motor and gear reduction. The coupling is a radial face type with a solid nonmagnetic metal diaphragm isolating the motor. Hence, the entire gear box, high speed as well as low speed gears and bearings, are open to space vacuum. The basic philosophy behind this configuration was not the achievement of long life, but the isolation of the organic components of the motor; i.e., brush debris, etc. At the same time, it avoided the complexity and failure hazard of a flexing bellows. Also, the magnetic coupling constitutes an excellent slip clutch (torque limiter), accurately presettable and without any sliding parts.

f. Motorized Switches - Motorized switches are of special interest in that they represent a variety of hermetically sealed motor which will remain in common usage, they will not accumulate any appreciable amount of running time even on a ten-year life spacecraft since their operating time is in the order of 200 miliseconds.

These motors are built into the switch mechanism and operate through a gear train. Series wound dc motors are usually used for this purpose for optimum torque versus speed characteristics. The motor never reaches a stable speed since it is switched off while still accelerating.

Such switches were installed on the Titan Missiles up to eight years ago and are activated periodically; however, during the ensuing period a significant number of switch failures have been experienced. Investigations have shown that commutator filming has been the cause. This is attributed to outgassing of either the lubricant or coil impregnant after years of service (Ref 1).

Another failure mode experienced during Apollo development was the glazing over of a commutator as a result of decomposition of F50 Silicone oil presumably due to arcing at the brushes.

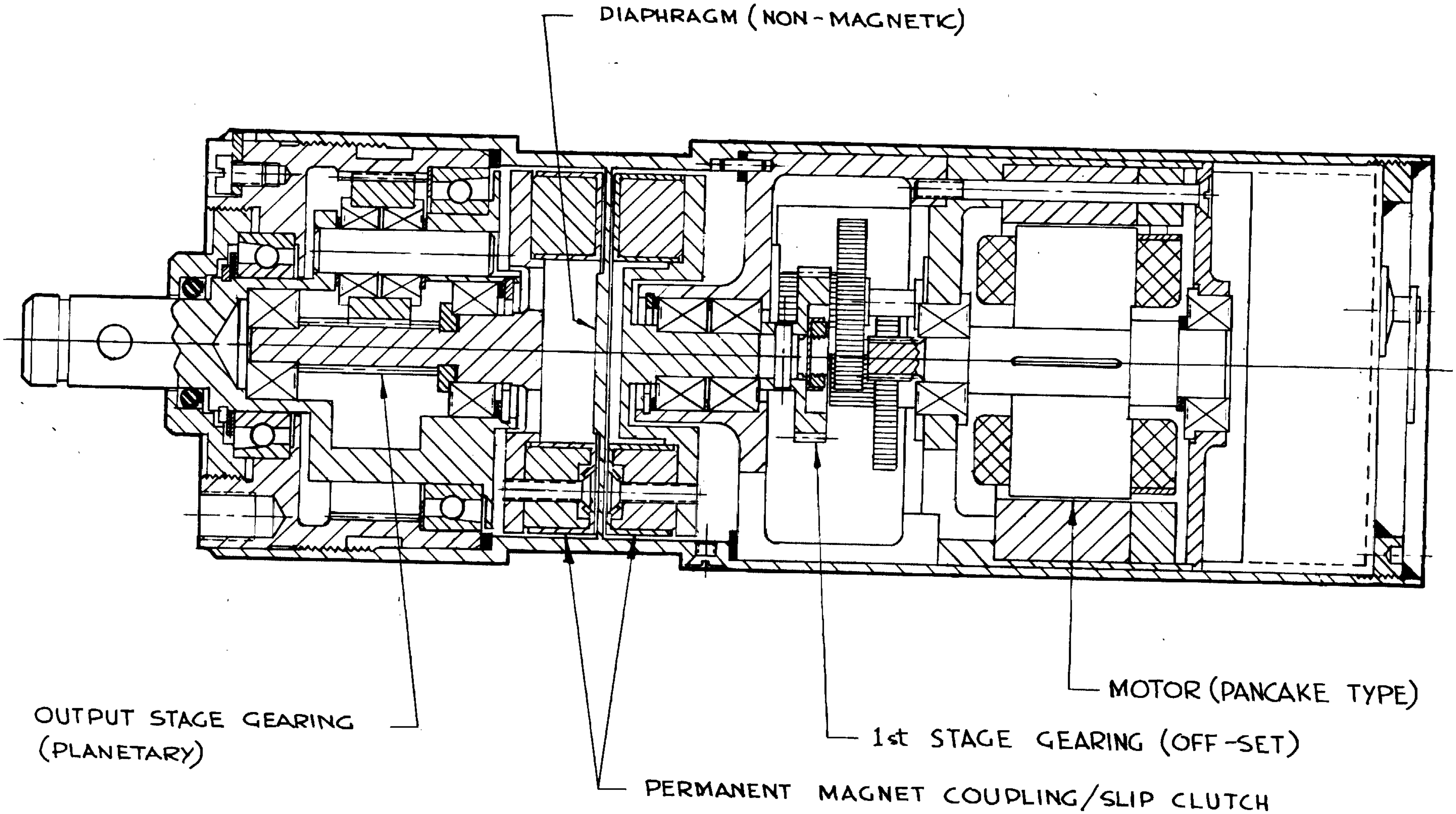


Figure 5 Hermetically Sealed Motor Using Magnetic Coupling

Courtesy of Singer Kearfott

These instances emphasize the extreme care that must be exercised in the selection of lubricant.

g. *DC Torquer-Brush Type* - DC torquers are merely a slow speed high torque version of the PMDC motor previously discussed. This is achieved by the use of comparatively many poles and large diameter air gap. As a consequence, the motor is of pancake configuration. It exhibits the same linear relationship between torque and current and between speed and torque.

For reasons that must be left to design treatises, they exhibit fast response, high linearity and resolution and are therefore ideally suited to closed loop servo-system. In fact, the vast majority of servosystems in spacecraft have utilized this form of servo-motor in preference to either ac or brushless dc (however, it is true that brushless dc motors are gaining ground in this area). Approximately 12% of motor applications have utilized this type of motor.

The torque motor differs from the conventional dc motor, not only in that is is of pancake configuration, but in the fact that it is almost always supplied frameless; i.e., without casing or bearings. It is often mounted onto a rotating assembly, thus avoiding the need for additional bearings. Gears are seldom used between load and motor and motor speeds are usually quite low. Hence, it has been possible to acquire continuous operation for several years in space vacuum, preimarily because brush velocities were low. Certain commutator lubes have facilitated this, pioneered by Ball Brothers' wet Vackote.

Typical examples are the OSO III to VII despin and elevation torque motors. Using wet Vackote and operating at 33 rpm's, these have operated faultlessly since launch. In the case of OSO III, launched in March 1967, the despin motor has executed 17.4×10^6 revolutions.

In the Apollo program, at least three torquers were used including:

1) Gimbal drive for the inertial measuring unit on the command module and LEM;

2) Service module camera;

3) LEM descent engine throttle valve.

On the Apollo Telescope Mount, a minimum of seven torquers are to be used for various control functions. On Viking, four torque motors are planned.

There have also been some noteworthy failures of torque motors, for instance, the momentum wheel drive motors on ITOS I and NOAA I, operating continuously at 150 rpm's in space vacuum. Each momentum wheel was provided with two identical motors for redundancy, but they both failed sequentially on both satellites. On ITOS I they lasted 14 months; on NOAA they lasted seven months representing 54×10^6 revolutions, respectively. Clogging of commutators by brush debris is the assumed failure mechanism--not simply a matter of brush wear.

It is probably unfair to classify these as failures in that the motors did accomplish up to seven-months (5100 hours) of continuous service, which was presumably their limit under the particular operating conditions. The motor manufacturer is of the opinion that an excess of oil was the fault. They have been superceded by brushless dc motors on subsequent ITOS satellites.

On the very same satellites, smaller higher speed torque motors, used in the tape recorders, were considered to have provided excellent service with 3800 hours of operation; however, these were located in the sealed interior of the tape recorder with a very beneficial relative humidity of 20%. Life test units have, in fact, accomplished 8000 hours of operation without failure.

(The life figures of actual accomplishments are being cited to give insight into current capability since there will be numerous instances in future "long-life programs" where this magnitude of actual operating capability will be more than adequate.)

Why should brush type torquers still be used when brushless types are readily available? This is often a matter of economy, not merely in regard to the motor itself, but of the commutating electronics which quite often are not supplied by the motor manufacturer. Also, the need for an aperature disc, photo diodes and light sources, alternatively resolvers or Hall Effect devices, to substitute for brushes, can trade-off disadvantageously, particularly when a vast catalog range of brush type torquers are available with no tooling or development costs.

Another important consideration is that the brush type dc torquer can be used at temperatures well below the capability of the electronics needed by the brushless dc motor.

Therefore, it would appear that, although brushes can be a potential hazard, the brush type dc torquer will remain an optimum choice in many future long-life applications.

No review of dc torquers would be complete without reference to the Mariner Mars 1971 gimbal actuator. It is a fine example of design simplicity (Fig. 6). It will be observed that a specially designed torquer is utilized having a radial face commutator, this having been necessitated by physical size limitations of the actuator. This torquer was designed to provide sufficient torque to permit the ball screw to be operated directly without need for gearing. The 94% efficiency of the ball screw was therefore not prejudiced. Feedback sensor is conveniently located inside the ball screw.

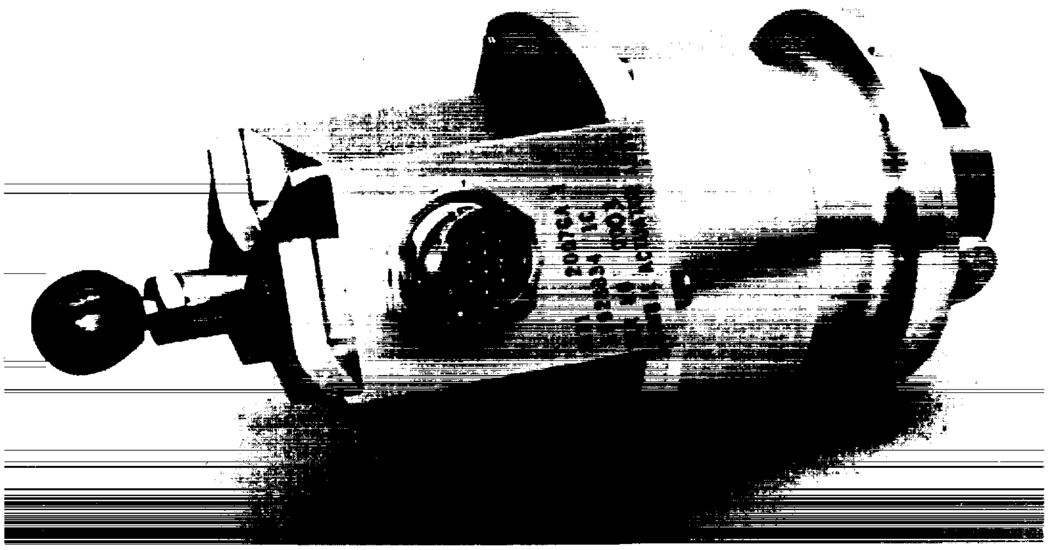
h. Induction Motors - There are many varieties of induction motors; single-phase, two-phase, three-phase, servo-versions, high speed conventional types, low speed pancake types, and canned versions. All are used in the space program. Each one was selected because of specific attributes which have proven advantageous in comparison to other types of motor.

Since (1) dc power is basic to all spacecraft, (2) an ac power source dictates the use of an inverter, and (3) since an ac motor has a less favorable power to weigh relationship compared with dc motors, it becomes even more evident that the adoption of ac motors must offer definite advantages, at least for certain types of application and environmental conditions.

Induction motors offer a compromise between the hysteresis synchronous and the brush type dc motor in that they are normally designed to provide only a small variation in speed over their full operating torque range.

This normal type of ac motor is available for either single-, two-phase or three-phase supply. The single-phase arrangement should be avoided since it merely utilizes a two-phase motor with a capacitor to produce the second phase with considerable loss of efficiency especially if the capacitor has to be sized for starting conditions.

(a) ASSEMBLED ACTUATOR



(b) CUTAWAY VIEW

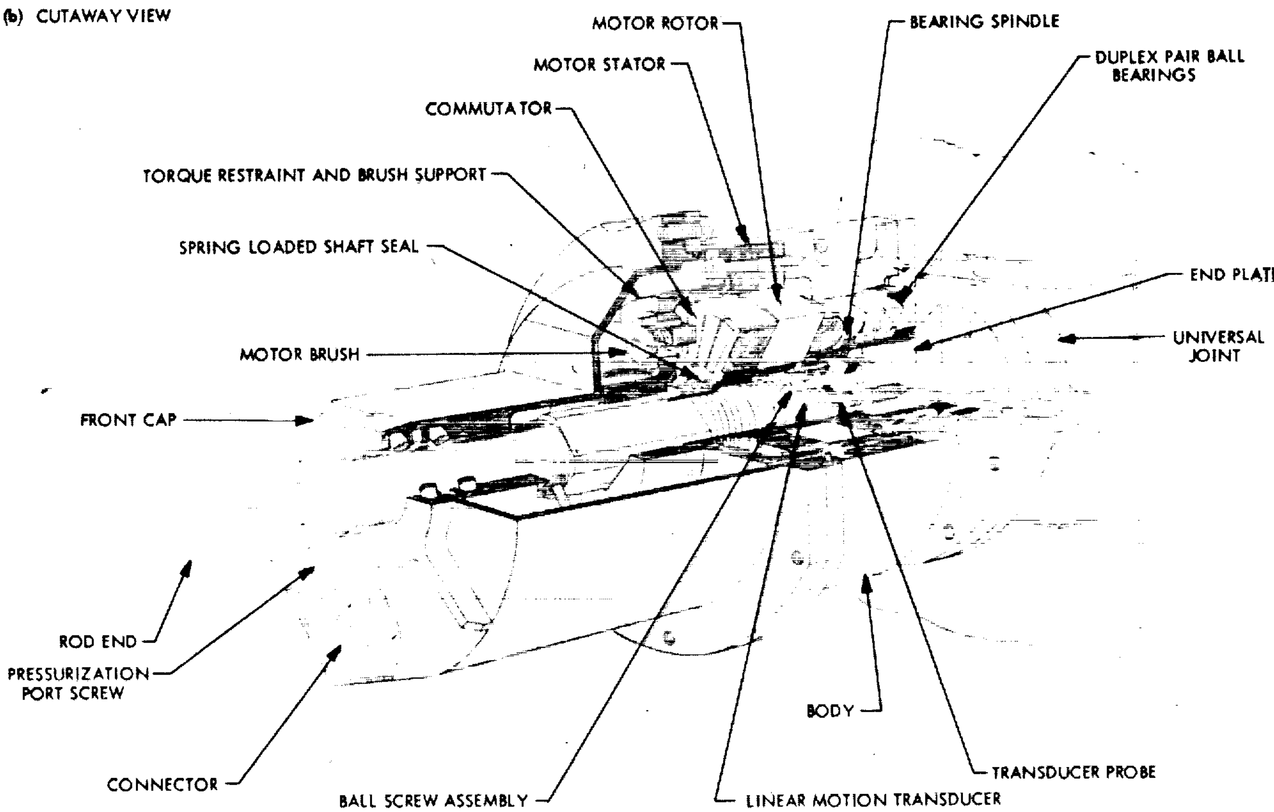


Figure 6 Mariner Mars '71 Gimbal Activator
Courtesy of Jet Propulsion Laboratory Pasadena, California

An illuminating example of the use of this type of motor is in the Horizon Scanner as used on over 300 Agena vehicles. Early models used a conventional high speed induction motor which necessitated the incorporation of reduction gearing to drive the scanning prism. For greater life and reliability a direct operating (without gearing) pancake induction motor was substituted in later versions (Fig. 7).

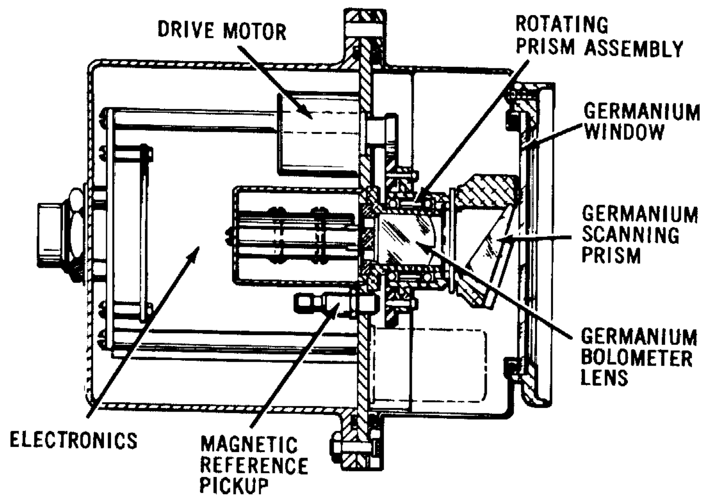
Another important application of the pancake type of inside/out induction motors has been for Reaction Wheels on many satellites such as NIMBUS, OAO, OGO, POGO LES, VELA. They have provided years of continuous and satisfactory operation at speeds in the order of 1200 rpm. It will be noted that the unit is completely sealed and pressurized. See Fig. 8 for a typical installation. It must also be observed that alternative designs of Reaction Wheels using similar but more compact configurations and vacated instead of pressurized have also provided years of satisfactory service.

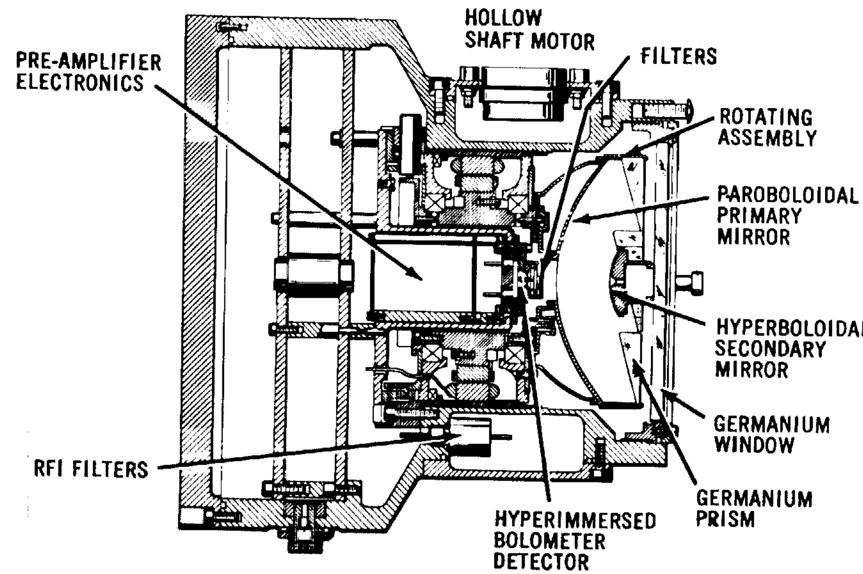
Two-phase and three-phase motors are of equal efficiency and power to weight ratio; the choice of which to use could be a matter of inverter preference or simply a matter of availability.

Induction servo-motors are of two-phase construction. They are designed for higher slip levels and provide a linear speed torque characteristic very similar to a dc motor. However, the ac device operates at high speed through a gear head, without brushes; whereas, the dc torquer avoids the gearing but does require brushes; It is a trade-off situation dependent on application requirements (duty cycle and life, etc) versus environmental constraints. The overriding possibility is that the final selection may be dictated simply by the geometry of the motors, pancake versus conventional, one or the other being impossible or inconvenient to incorporate into the available envelope.

A very successful application of a two-phase, 400 Hz ac servomotor is to the Scan Platform Actuator on Mariner, located in a sealed enclosure. That on Nimbus I for driving the Solar array failed after one month due to excess bearings temperatures and lubrication failure. Larger motors proved successful on subsequent Nimbi.

THEN AND NOW — Capability growth of conical scan horizon sensors





Parameter	Project MERCURY Sensor	Present-Day Sensor
Optical System	Refractive	Reflective and refractive for highest transmission efficiency
Spectral Band	2 - 22 microns	14 - 16 microns (CO_2 band)
Rotating Assembly	High-speed motor and reducing gears	Low-speed motor, no gears, all materials matched for compatible coefficient of expansion
Reliability	Commercial components	High-Rel components
Ambient Temperature Operating Range	+20°F to +120°F	−30°F to +165°F
Attitude Output Redundancy	None	Redundant outputs
Protection Against Signal Loss	None	Monitor circuits inhibit erroneous outputs
Instrument Accuracy	±1°	±0.1°
Possible Error due to Cold Clouds and Other Earth Radiance Variations	3° - 6°	Less than ±0.05°, due only to earth radiance variations
Errors Caused by Sun in Field of View	Up to 7°	None
Electronic Noise in Sensor Output	±0.2°	±0.02°

Figure 7 Induction Motor - Product of Evolution

Courtesy of Barnes Engineering

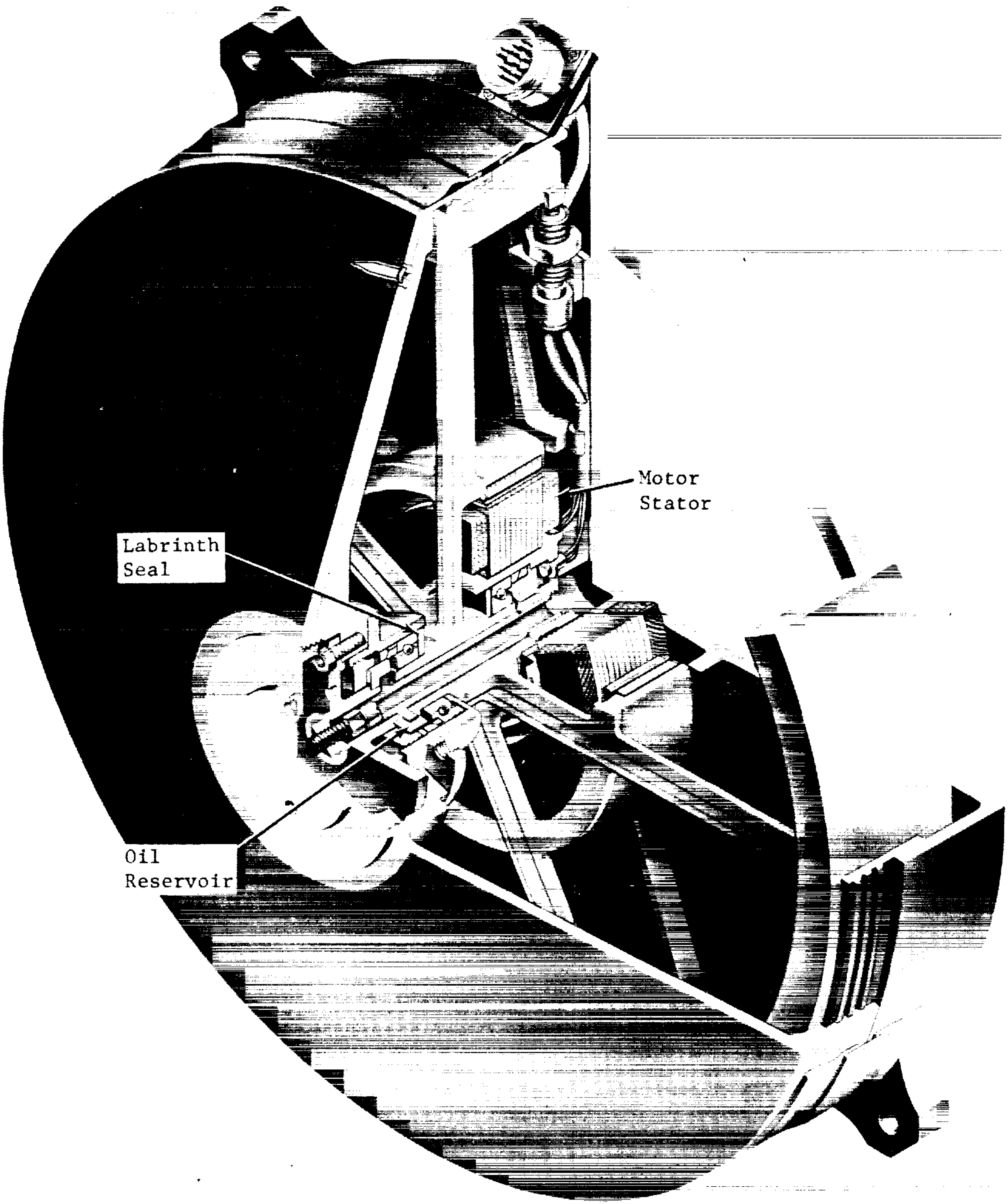


Figure 8 Pancake Type of Induction Motor

Courtesy of The Bendix Corporation

The induction motor is also the logical choice for canned pumps wherein the stator is hermetically sealed using a tubular membrane between stator and rotor. A typical example of this is the Skylab Airlock Coolant Pump. This has tested satisfactorily throughout 5000 hours.

Induction motors are the usual choice for wet pumps (where the motor interior is flooded) and completely submerged pumps (such as the Hydrogen Fuel Cell Pump on the Apollo Service Module, which is a three-phase unit.)

For pumping of corrosive liquids the use of a magnetic coupling is a preferred solution utilizing induction motors. This could also tradeoff advantageously compared to the use of a canned motor for long life noncorrosive liquid pumps since viscous drag is avoided and a less powerful motor can be used; however, there are twice as many bearings involved with this arrangement.

i. Brushless DC Torquers - Two basic types of brushless torquers are in use: the continuous rotation type, and the partial rotation type. Both are servomotors and both compete with the brush type torquer. All are of pancake configuration and are usually applied directly to the driven shaft.

Of the 26 (8.6%) spacecraft applications of brushless dc motors, approximately 62% are of the partial rotation type. This partial rotation type involves no (electronic) commutation; and is extremely simple, merely involving two components, the PM rotor and the wound field. They trade-off considerably heavier than the brush type torquer; but they neither exhibit any friction or ripple nor do they involve brushes. A typical application of this type of motor is for pitch, yaw and roll control of Earth Sensors, such as used on OGO, POGO, LES and VELA; these operate on flexures instead of conventional bearings.

The eight remaining applications utilize the continuous rotation type of brushless dc torquer. Since these have all been specifically selected for longevity reasons, details are being presented in Table 1. It will be noted that they are all resolver commutated and that a redundant resolver was incorporated. The arrangement illustrated in Fig. 9 is designed for a 5 year life capability on IDCSP/A and Skynet. The totally redundant unit, as used on Telsat, is shown in Fig. 10.

It will be observed that many of these are for despin functions on the more recent long life communications satellites, having been selected in preference to dc torquers.

Table 1 Brushless DC Torquers in Spacecraft Use

Satellite	Function	Operating Torque OZ Ins	Stall Torque	Normal Speed rpm	Size	Life Years	Manufacturer & Model No.	Comments
DSCS 2	Solar Array Despin		180	Zero to 180	5-1/2" dia x .44		Westinghouse	Embodies redundant windings, resolver commutated.
IDCSD/A	Despin	35 at 10.6 volt		100	4-5/8" dia x 3/4		Aeroflex TFR 47-IP	Resolver commutated.
HELIOS	Antenna Despin	160/amp			4.95" dia x 1.17		Sperry	Inside-out configuration, resolver commutated.
SKYNET	Antenna Despin			76 to 105	4.75" dia	5-yr requirement	Aeroflex TFR 47-SP	Resolver commutated.
TELSAT (Two)	Antenna Despin & Dynamic Balancing	58 at rated speed		120			Sperry	Embodies dual redundant winding and dual resolvers inside-out configuration.
INTELSAT IV	Despin Platform	.85 ft lb per amp	1.8 ft lbs	60	4.624" dia	7-yr requirement	Aeroflex TFR 47-9P	Embodies redundant windings, resolver commutated.
ITOS II & III	Momentum Wheel Drive			150			Macbar	Resolver commutated.
LEM	Radar Range Shaft Actuator		175	105 (NLS)	4.5" dia x 1.18		Bendix 1877055	Resolver commutated.
LEM	Radar Range Shaft Actuator		325	105 (NLS)	4.5" dia x 1.75		Bendix 1877056	Resolver commutated.

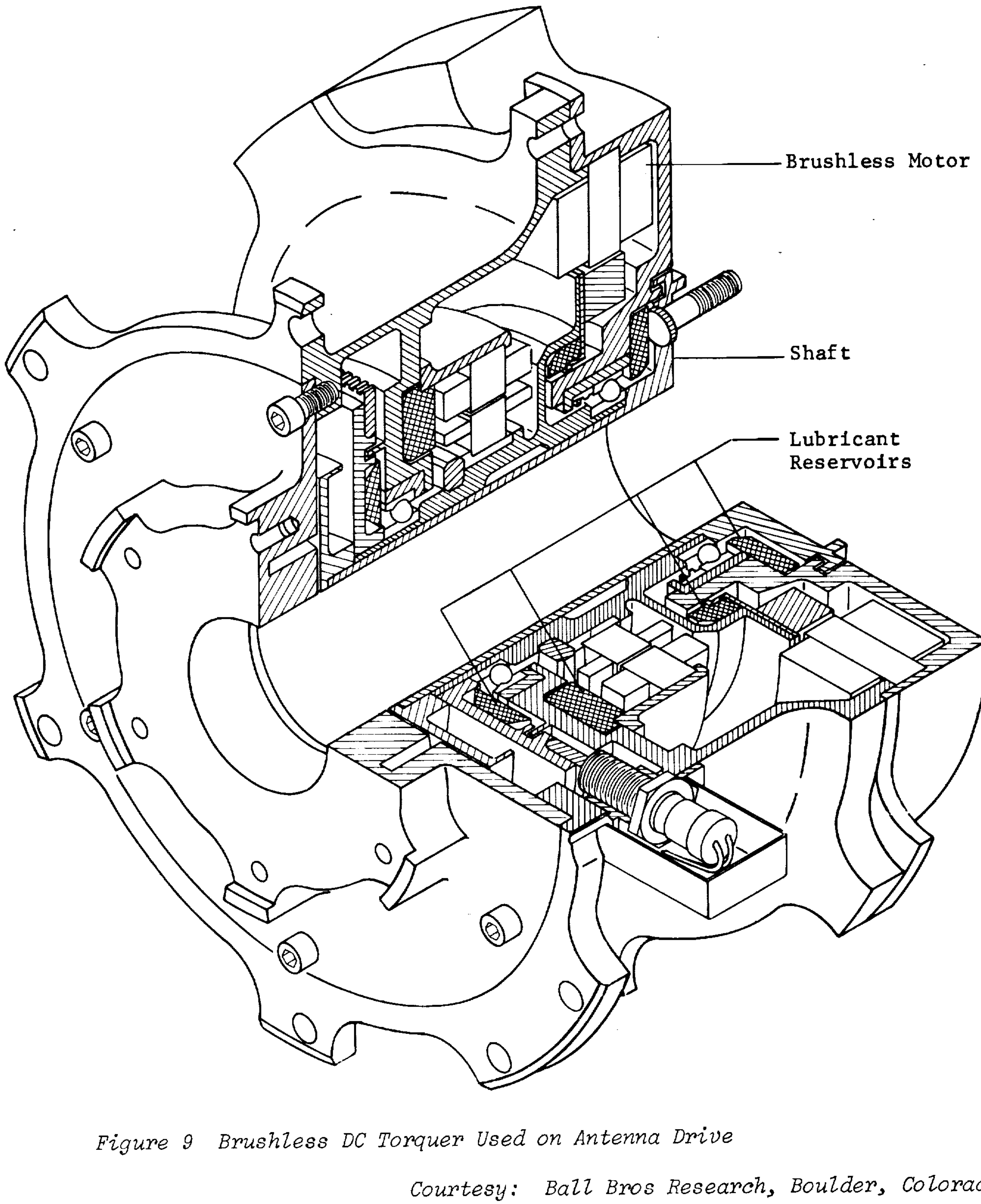


Figure 9 Brushless DC Torquer Used on Antenna Drive

Courtesy: Ball Bros Research, Boulder, Colorado

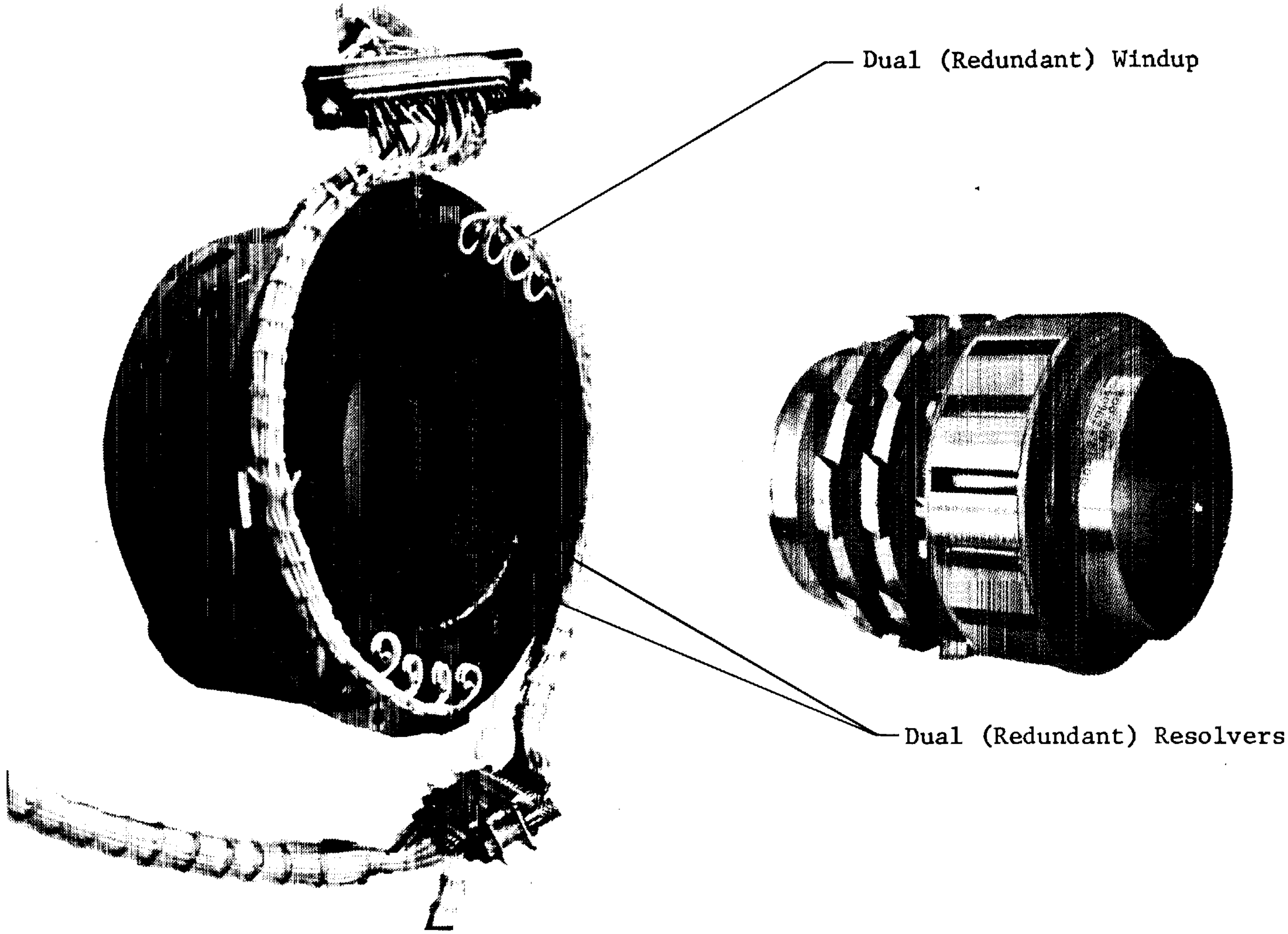


Figure 10 Brushless DC Torquer with Redundant Windings & Resolvers for Telsat

Courtesy of Sperry Electronic Component Plant
Durham, North Carolina

In comparison with dc brush type torquers, the dc brushless torquer can offer approximately equal performance for the same weight; however, the necessary resolver and electronics constitute a weight, complexity and reliability penalty which has to be traded off against the simplicity, yet unreliability of brushes.

Hall device commutating represents another trade-off feature which may be used advantageously in limited temperature environments and is under consideration for future tape recorders to eliminate the speed reduction belts and pulleys necessary with Hysteresis Synchronous motors. However, the commutating electronics is a complection which must be traded off against the mechanical complexity In regard to optical commutating, manufacturers seem to be limiting this to small high speed motors.

An advantage of the brushless motor is that its heat generating portion does not rotate; hence, heat removal is optimized. Bearings do not have to carry this heat as in the case of brush type motors.

j. *Variable Reluctance Stepper Motors* - These are usually used for their small step angle characteristics; however, they do not offer the (deenergized) detent torque feature of the PM Stepper.

One of the most interesting applications of this type motor to date, illustrating long life capability, is the system for antenna positioning on Intelsat IV. In this case, it drives a jackscrew via a 4:1 gear ratio. It normally remains completely dormant, but must remain operable should it be required to change the satellite's position at any time during a 7 year life.

Bearings are burnished with MOS_2; gears and jackscrew and gears are lubricated with Lubeco 905 dry film lube.

k. *Latching Stepper Motor* - This type of motor is unique to one manufacturer, Abrams Instrument Corporation. It has been used successfully in at least 13 spacecraft applications. It is currently being incorporated into two camera installations on the Apollo Telescope Mount.

Unlike the PM Stepper, which exhibits a dormant magnetic detent torque of approximately 10% of its operating torque, the Abrams motor incorporates mechanical latches. These latches are released by the initial motion of the rotor, continued motion of which indexes the output shaft, whereupon the latches are redeployed.

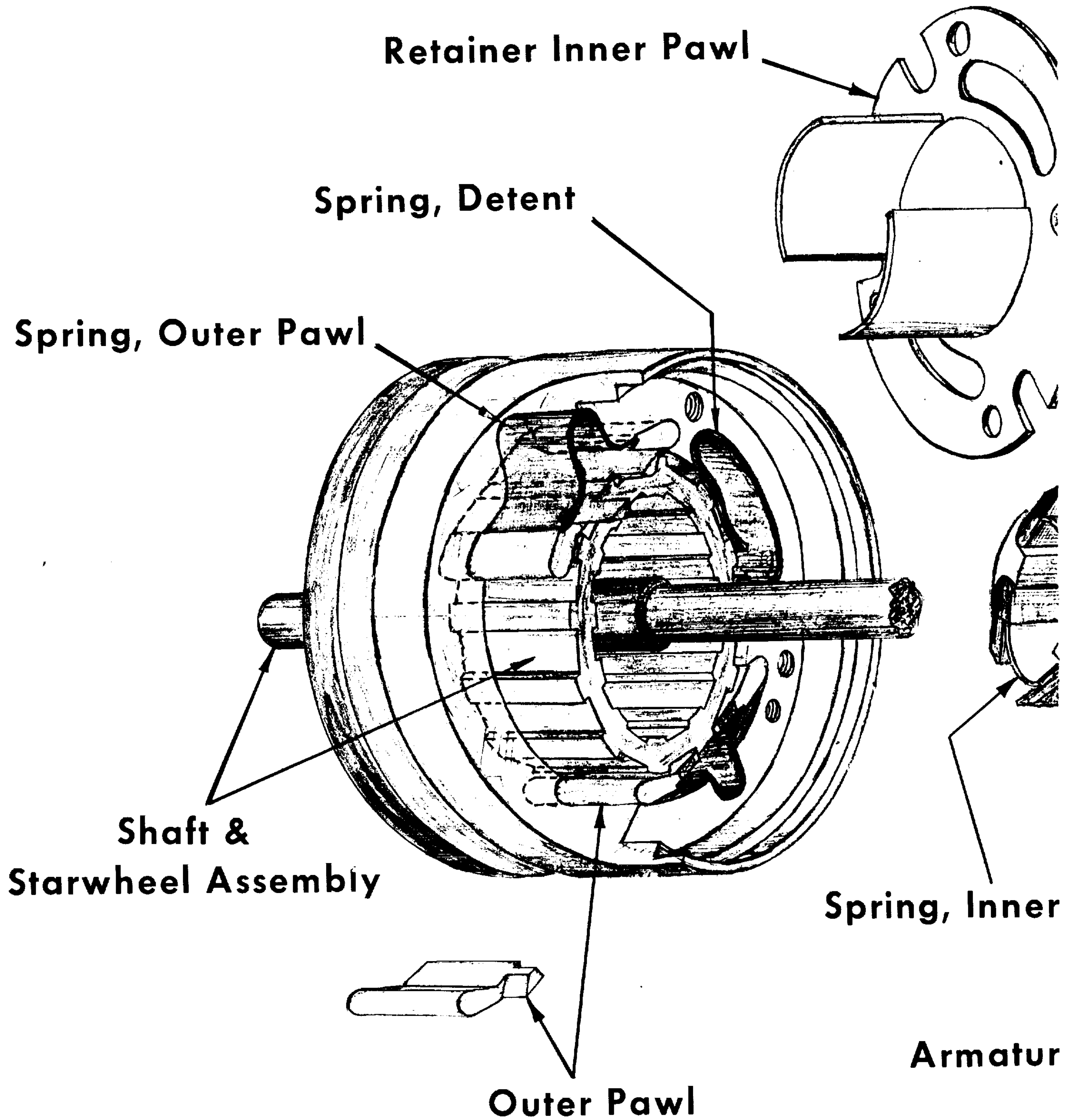
On OAO II, the mechanisms were gold-plated and the plain bearings in the armature (where it rotates on the output shaft) were supplied in Rulon instead of the usual dry film lubricated brass. These are identified in Fig. 11.

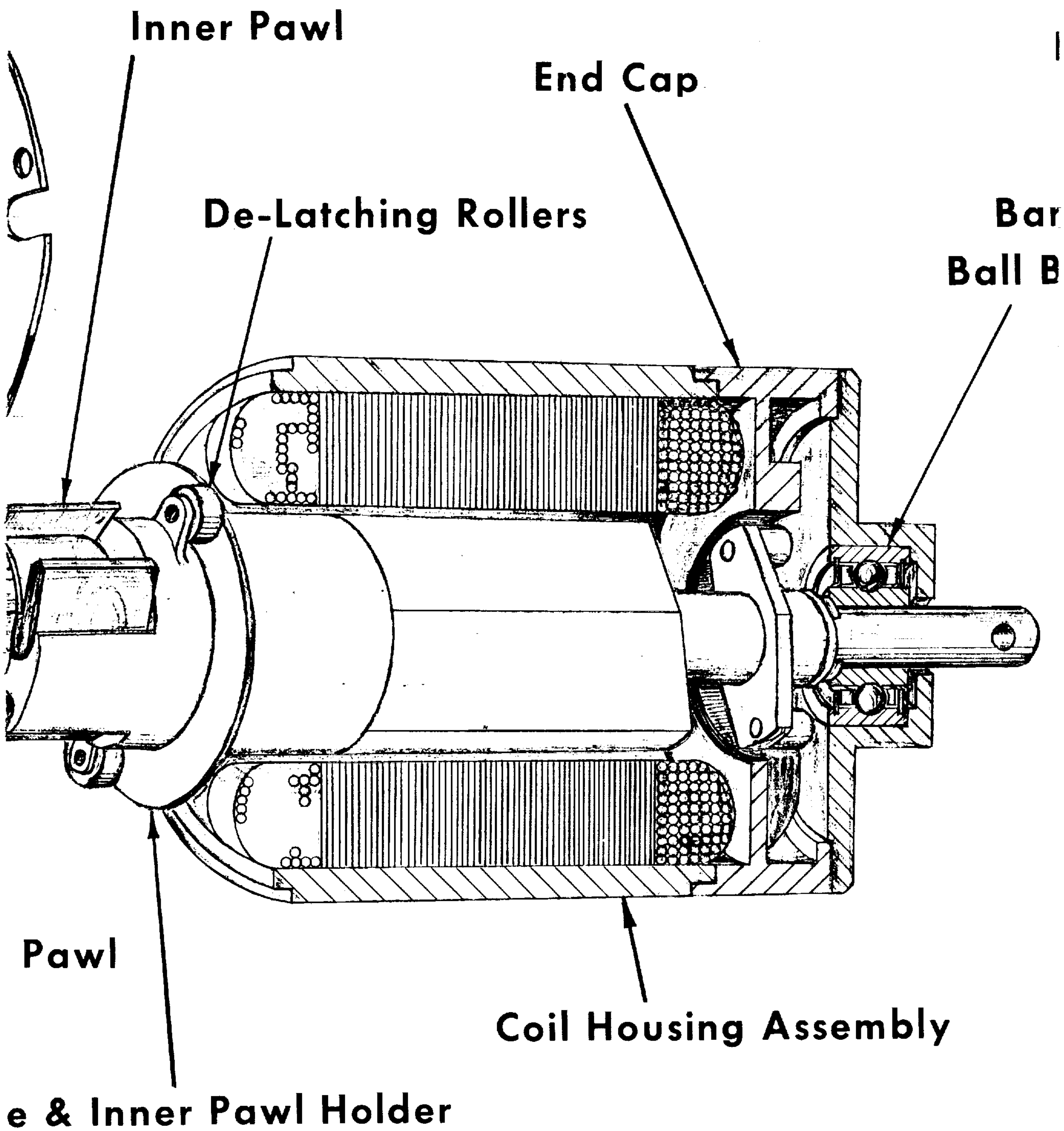
Caution: This motor may be unsuitable in situations where the motor shaft is subjected to an appreciable live load. During the return motion of the rotor, there are a few microseconds when the latches are not holding the output shaft and under test it has been shown to slip a step.

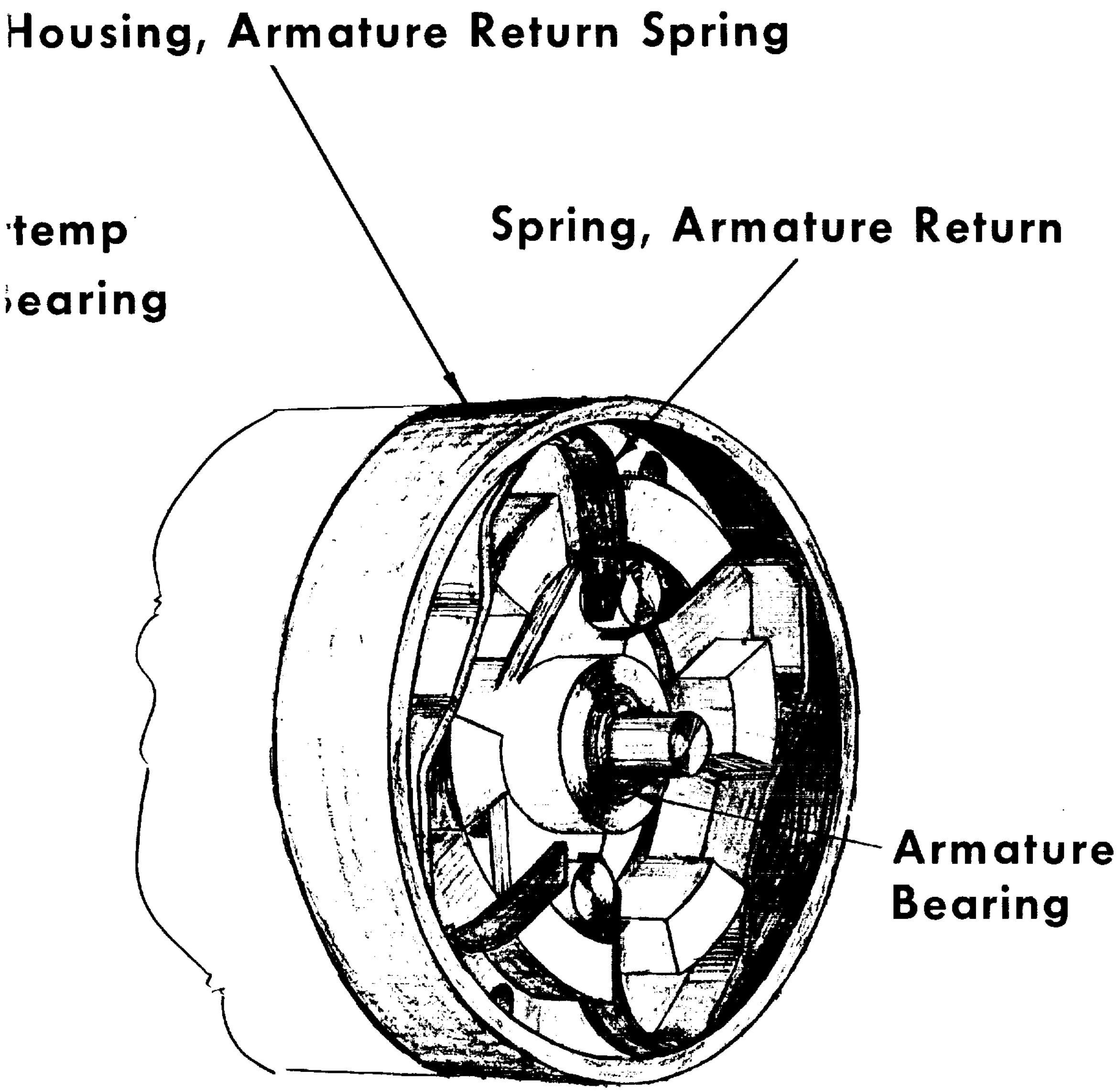
ℓ. *Brushless DC Motors* - These differ from brushless dc torquers in the same fashion that brush type dc motors differ from brush type dc torquers; i.e., they are of conventional (cylindrical) configuration and are supplied as an integral working package complete with bearings and the electronic and photoelectric components for commutating, or in some instances, Hall effect devices are used.

While these motors are very efficient, in the order of 75% to 85%, they are nevertheless considerably heavier and more bulky than the equivalent brush type motor. However, these motors are still in evolution; and, thus, this situation should improve.

They compare favorable with the induction motor plus its invertor and are eminently suitable for extended usages, such as for driving fans. It is in this role that they have been used in the environmental systems on Gemini and Apollo and for space suites. All of these have used the photoelectric method of commutation.


Retainer Inner Pawl
Spring, Detent
Spring, Outer Pawl
Shaft & Starwheel Assembly
Spring, Inner
Outer Pawl
Armatur


Inner Pawl
End Cap
De-Latching Rollers
Bar
Ball B
Pawl
Coil Housing Assembly
e & Inner Pawl Holder



End View-End Cap Removed
Showing Armature Return Springs

Figure 11 Abrams Digital Stepper Motor Model DMA

B. GUIDELINES FOR LONG-LIFE ASSURANCE

Ungeared induction motors and PM stepper motors are currently registering up to five years of continuous operation in hermetically sealed environments at 1500 rpm. In laboratory conditions, eight years have been accomplished. Other high speed ac motors (brushless) derated to 3000 rpm have achieved up to 3½ years continuous operation with and without gearheads, hermetically sealed or with molecular shaft seals. In all cases, fluid lubes were used.

Very slow speed brush type motors at 30 rpm are still operating satisfactorily after 4.6 years, but other brush type motors operating at 150 rpm failed after approximately 7 months from either brush, commutator, or lubrication problems. Commutator diameters were almost identical in both instances; thus, the brush linear velocity on failed units was five times that of units still operating. Wet lubes were used in all motors.

Specialty applications of motors to pumps and fans are covered in appropriate sections chapter V.

Brushless despin motors at 100 rpm, still operating after 3½ years, are destined for a 7-year life.

Apart from the brush problem of dc motors, bearings and their lubrication constitute the primary life limiting feature of motors in satellite applications. The following guidelines concentrate on means to enhance the current state-of-the-art for assurance of optimum life by "designing in" the necessary reliability and life.

1. Application and Design Guidelines

1) Employ minimum gearing between motor and load. Low gear ratios require less gears and bearings and allow larger gears that have more durable gear proportions.

2) Use the largest bearings consistant with design constraints. Larger bearings are: better resistant to brinelling and false brinelling, have lower contact stresses (longer fatigue life), and are stiffer (higher natural frequency).

3) Use low speed, multipole, pancake motors in preference to high speed motors. The shorter, bearing span in the motor provides a higher natural frequency to the rotor shaft. Also, better heat transfer from winding through the casing to the heat sink (bypassing gearing) is provided. They can be mounted more solidly, reducing vibration amplification.

(Note: It should not be construed that high speed of itself is a hazard. But, as long as speed reduction is involved, it should be accomplished with the minimum feasible gear ratio.)

4) Use wet lubricants in preference to solid lubrication (ambient temperatures permitting) because:

a) Boundary lubrication with wet lube (or grease) is preferable to that obtained with solid lubrication.

b) Wet lube is a better heat transfer medium, allowing bearings to dissipate conducted or generated heat more readily.

c) Ability of wet lubricants to operate in EHD or partial EHD regime. Life is limited only by lube film endurance; the wear is insignificant.

5) Use a channelling type grease when using grease for rolling contact bearings. This allows the balls to form a channel which the consistance of the grease is designed to sustain. Friction is thereby minimized. Oil will bleed into the ball track from residual grease.

(Note: Grease may be preferable for rolling contact bearings instead of oil in order to acquire higher viscosity in low speed gearhead (or motor) bearings. This permits operation in a preferred lubrication regime.)

6) For motors that will be repetitively subjected to earth environment and space vacuum; (i.e., shuttle operation) the following policies are recommended:

a) Use hermetically sealed motors and gearheads when the output shaft speed is 200 rpm* or less and wet lube is feasible. This excludes contaminating and humid conditions when in the earth environment. It avoids outgassing contaminant from the motor and prevents lubrication loss by evaporation when in space.

b) When a brush type dc motor is employed above -10°F, and the service life is less than 400 hours, utilize a moisture content in the encapsulated gas of 2 grains of H_2O/lb of air in conjunction with high altitude carbon-graphite brushes. Moisture acts as a lubricant for carbon-graphite.

c) In lieu of Item b) above, consider the use of a direct acting frameless dc torquer, with or without brushes. Use a molecular shaft seal. The advantages are simplicity, sturdiness and elimination of high speed bearings and gears.

d) For high speed motor applications (no gearhead), use molecular shaft seal and lubricant reservoir system. This type of seal introduces no torque penalty.

e) On slow speed, high torque applications, elastomeric shaft seals may be feasible to enable the motor to be pressurized with an inert gas (i.e., power hinges on booms). Elastomeric seals will exclude contaminating and humid conditions when in the earth environment. They also avoid the complexity of hermetic sealing and prevent lubrication loss by evaporation.

f) Use either metallic or viton bellows for sealing motor actuators with linear output. Pressurize interior to ½ atmospheres with inert gas. This will exclude contaminating and humid conditions when in earth environment. It will prevent lubricant loss by evaporation in space.

7) For long brush life on dc torquers, utilize cartridge type brushes instead of the usual cantilevered spring type. Longer brush lengths can be used to accommodate the higher brush wear rates in vacuum.

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*This speed limit is somewhat arbitrary; it is assumed that below this speed, torque levels should be high enough to handle the additional load of the hermetic sealing without serious penalty.

8) Avoid the use of silicone oil in brush type torquer applications. If oil migrates to the commutator, arcing may cause decomposition and result in a film of silicone on the commutator.

9) When using silicone lubricants, pay special attention to the minimizing of oil migration. Barrier films may be used. A preferred alternative is teflon components sandwhiched between bearings and housing. Employ teflon coating of molecular shaft seal surface.

10) When using molecular seals and reservoirs, locate reservoirs immediately adjacent to each bearing such that oil migration is facilitated. Bearing lubrication does not rely entirely on vapor condensation.

11) Also, locate a reservoir adjacent to the motor winding to quickly generate an oil vapor following inoperative periods at low temperature.

12) When using molecular seals and reservoirs, prewet all interior surfaces of the motor with a film of oil to avoid unnecessary loss of reservoir content.

13) When using molecular seals and reservoirs, incorporate an air filter into the body of the motor to bypass the majority of air flow from the molecular seal. Filter must provide high laminar flow and low molecular flow compared with the molecular seal. The air filter avoids particulate contamination of bearings and molecular seal.

14) When using molecular seals and reservoirs, in conjunction with a brush type motor, special care must be exercised to maintain an optimum film thickness on the commutator. An excess of oil may be as detrimental as an inadequacy. The optimum film thickness will help optimize brush wear, minimize arcing and avoid lubrication degradation.

15) When using grease lubed ball bearings, select cartridge type bearings (this does not apply to angular contact bearings). These are longer than standard and permit a larger charge of grease.

16) When using shielded bearings, insure that shields are leaktight where they interface with outer race to prevent creepage loss of lube from bearing.

17) For EHD or partial EHD operating conditions, use precision bearings ABEC 7 or ABEC 9 grade with following overriding tolerance and finish requirements. These are prerequisites for proper EHD or partial EHD operation. When using bearings to the below tolerances, housings and shafts must be manufactured to similar accuracies of peripheral roundness in order to maintain bearing accuracy after installation.

- Race groove, OD and peripheral roundness and eccentricity:	0.000010 in. max TIR
- Cross-Race departure from constant radius (in ball contact region):	0.000010 in. max
- Surface finish - balls:	1 microinch rms
- Surface finish - raceways:	2 microinch rms
- Visible damage, asperities, blemishes (at 40X) on balls or in ball path of raceway:	None
- Ball roundness:	0.000003 inches TIR
- Ball accuracy within any bearing or batch:	0.000005 inches

18) Use 52100 consumable electrode vacuum melted alloy steel bearings except when a corrosion hazard exists, otherwise utilize 440C melted in the same manner. The 52100 exhibits fewer inclusions even when consumable electrode vacuum melted. It also has minimum dimensional change with time.

19) Use ABEC 7 quality bearings for maximum assurance of quality, quality control, precision and cleanliness. When greater clearances than normally associated with the ABEC 7 grade are essential to accommodate for dry film lube thickness, the same quality precision and quality control provisions should be maintained. The same applies to instances where ABEC 9 tolerances may be necessary in order to meet preload or stiffness requirements.

20) Use porous, vacuum oil impregnated ball separators, either laminated phenolic, nylasint or porous polyimid. Emphasis should be placed on using the latter material which, although of recent origin and not cataloged, appears to offer considerable advantage. Polyimid has higher porosity than laminated phenolic, and is more readily (and accurately) machined than nylasint.

21) Exercise extreme care when cleaning and vacuum impregnating. Use procedures developed by NASA Goddard as a guide which are given in Section E. These ensure thorough removal of preservative oil applied by the bearing manufacturer, satisfactory wettable surfaces and maximum oil capacity of the porous material.

22) To accommodate extremes of temperature, anchor the rotor longitudinally with one bearing. Accommodate differential thermal expansion and contraction by allowing axial float on the other bearings. This will avoid excessive thrust loads on bearings due to thermal expansion and/or contraction.

23) To prevent brinnelling of bearings and housing, avoid the arrangement where both bearings float, except under the mildest vibration and temperature environments.

24) Utilize housing materials (berillium or titanium) with a temperature coefficient of expansion compatible to that of the rotor, when operating temperatures vary widely. This will:

    a) Minimize change of bearing accuracy, dimensions, friction and loading with temperature change;

    b) Permit minimum clearance to be used on the floating bearing, thereby avoiding false brinnelling between housing and outer race;

    c) Permit smaller air gap (better motor efficiency);

    d) Minimize thermal strains between stator and housing.

25) Alternately to 24) use "compatability" bushings in the housing for the same reasons; except the last.

26) Use symmetrical bearing housing geometry to avoid distortion of bearing bores (and of the installed bearings) due to assymmetrical temperature expansions or contractions.

27) Close control of the housing/bearing and shaft/bearing fits are essential. The degree of interference changes bearing clearances and contact angle; and in the case of duplex preloaded bearings, spring rate is modified. Hence, when these parameters are important, bearing bores and OD's should be custom coded so that they can be matched to housings which provide the required fit.

28) It must not be assumed that the catalog recommended fits for a particular bearing will suffice when minimal lubricant films are involved. This is particularly true where a temperature differential may develop between inner and outer race. Clearance fits may be necessary under room temperature assembly conditions in order to acquire optimum running conditions at operating temperature. Material selection for shaft and housing may have to be determined on the basis of these considerations. Fits must be individually computed in such circumstances.

29) When subject to extreme vibration levels and duration, design the rotor assembly such that its resonant frequency is substantially above the maximum input frequency. Using a duplex angular contact bearing at one end of the rotor and a radially preloaded bearing at the other end can facilitates this. Match bearing spring rates as closely as possible. These approaches will minimize amplification of input G levels to ameliorate false brinnelling hazard to bearings.

30) Alternatively to the above 29), the use of vibration isolators should be investigated to ameliorate false brinnelling hazard to bearings.

31) Mechanical worst case analysis shall be invoked to give maximum visibility to the combined effects of environment extremes in conjunction with manufacturing tolerances (including eccentricities, nonparallelisms) and the effects of operating temperature. Such analysis shall constitute a permanent part of the design record and be subject to approval by the contractor. Worse-case analysis will:

a) Provide a cost effectivity measure to minimize development and qualification problems;

b) Assure that manufacturing tolerances are determined systematically, nor arbitrarily; i.e., their full ramification on motor or bearing configuration being duly evaluated by means of the worst-case analysis;

c) Force consideration of the effects of temperatures, systematically evaluated and not left to intuition;

d) Insure close control on the tolerances and factors which influence bearing misalignment.

2. Process Control Guidelines

1) Preserve utmost cleanliness of bearings. Demand whiteroom assembly and packaging to Federal Standard 209a, Class 100. Also, require ultrasonic cleaning of parts followed by a wash and particle count. Cleanliness is essential for low friction, low wear and maximum life, especially for high speed applications and EHD operation.

2) Individually inspect each bearing and its components for accuracy and maintain a permanent record of the following parameters:

a) Raceway peripheral roundness and wavyness;

b) Eccentricity of ball path to OD or ID;

c) Cross-race profile at ball path;

d) Surface finish of balls and raceways;

e) Boundary dimensions;

f) Ball retainer dimensions including the bearing land diameter and roundness where it engages raceway;

g) Electronic (noise) analysis of the assembled bearing as a further check on integrity of running surfaces. See Reference 13;

h) Breakaway and/or running torque.

3) Inspect all raceways for imperfections using a 40X minimum magnification. There shall be no evidence of inclusions, comets, furrows or pits in the ball path. These will help optimize conditions leading to prolonged lubricated life.

4) Inspect ball retainers not only for accuracy and cleanliness, but also for complete freedom from burrs. Burrs constitute a serious contamination hazard in that they can work free during the life of the bearing.

5) Lubricant or preservative should be applied to bearing using a 30-micron absolute filter on the syringe* (i.e., ASTM 325 mesh screen) for added assurance of lubricant cleanliness.

6) Contamination level of lubricant to be strictly observed and checked by QC prior to use. For greases, the following cleanliness level shall be observed:

   - In any m/litre of grease there shall be no more than 1000 particles between 25 and 75 microns and no particles in excess of 75 microns.†

7) For oil lubricants invoke National Aerospace Standard 1638, Class 0, for added assurance of lubricant cleanliness.

8) Each batch of lubricant shall be individually tested to insure that properties are within specification. This shall include evaporation rate, viscosity, vapor pressure, pour point and viscosity index as a minimum. This will avoid the wide variation of properties for which different batches of the "same lubricant" are notorious.

9) Bearings utilizing dry film lubricants shall be run-in to develop a tenacious ball track. Following this operation, the bearing shields shall be removed and debris resulting from the run-in shall be removed. This run-in may be accomplished prior to assembly into the motor whenever ball loading can be accurately simulated as in the case of angular contact bearings. This should alleviate contamination as a failure mode.

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*These are tentative suggestions pending further investigation into the topic; see recommendation of Section 2c.

†This cleanliness level is being proposed for Revision B of MIL-G-81322 and represents optimum cleanliness, such greases being made under white-room conditions.

10) Bearing installation in housings shall be conducted in accordance with a written procedure under rigid inspection. This process is critical to proper bearing operation and must be performed with care and precision. The written procedure is to insure that the same (satisfactory) procedure is followed on subsequent production lots.

11) When ultrasonically cleaning assembled bearings, they shall be suspended and not allowed to rest on the bottom of the tank. This avoids false brinnelling hazard.

12) White room assembly conditions (properly supervised and controlled) should be regarded as essential for the more critical motor applications, particularly those involving EHD lubrication. Laminar flow bench assembly and test should be adequate for motors of less criticality; i.e., of low speed or where brushes will quickly generate debris. However, items such as bearings, which are white room processed, must be depackaged on the laminar flow bench.

13) Inspect commutator for freedom from burrs to eliminate brush wear hazard and possible bearing contamination hazard.

3. <u>Test Guidelines</u>

1) There is no point in qualifying a purely nominal configuration when it is the extremes of manufacturing tolerances which present the maximum hazard. A realistic qualification policy is required. Multiple qualification test units shall be employed, (in a cost effective manner) embodying the worst case extremes of manufacturing tolerances such that qualification can be considered to cover any permutation of manufacturing variables.

2) Acceptance testing shall include an extended run-in test where load, environment and heat sinking is simulated.* This shall be followed by a room temperature baseline performance test. Performance, including winding resistance and bearing or bearing housing temperatures, shall be continuously monitored.

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*Usually of several hours duration, depending on service life and duty cycle. As much as 200 hours have been demanded; the longer periods pertain to geared head motors to detect infant mortality of the slower bearings.

This record shall extend to any prior wear-in or run-in process during manufacturing and shall evidence no perturbation from normality. Bearing noise should be intermittently monitored. The test provides an opportunity for "infant mortality" failure mechanisms to develop and any other abnormality trend to be detected.

3) Testing of bearings has been largely covered under the topic of Process Controls, where noise and breakaway torque tests are specified. Also recommended are stringent microscopic tests of individual components, which could also be regarded as tests.

   A test instituted for gyro bearings, measures surface wettability, assessing the surface for contaminations which negate the effectiveness of fluid lubricants. It is submitted that the wettability test be conducted either periodically or once per production run. This would assure cleanliness of processes, and check on housekeeping measures and white room atmospheres.

C. LIFE LIMITING PROBLEMS AND SOLUTIONS

As signified in the introduction, every motor application involves its own combination of environmental, functional and installational problems. It is most unlikely that any one particular motor will prove ideal from every aspect. Compromises will be necessary. The optimum motor for a particular situation is that which involves the least amount of compromise--and this will naturally involve economic considerations (such considerations, however, do not constitute part of this study). Meticulous trade-off studies will be necessary.

The information in this chapter is designed not only to assist in the selection of the most appropriate type of motor, but also in the evaluation and determination of detail design features.

In regard to the analysis of failure mechanisms: while it is important to know how to overcome problems after they have arisen, it is even more important to know how to design the product so that problems won't occur, i.e., to design reliability into the product. Section 2. below will be directed toward this end. Section 1. will reference appropriate sections of 2. whenever pertinent solutions are examined in greater detail.

1. Failure Mechanism Analysis

The only motor failure mechanisms on spacecraft reported to date have related to bearings and their lubrication, and to brushes and commutation.

This analysis has, therefore, concentrated on these failure modes. Table 2 delineates the types of failure mechanism, cause and solution, associated with each failure mode.

Table 2 FAILURE MECHANISM ANALYSIS

MOTORS AND/OR MOTOR GEARHEADS - BEARINGS

FAILURE MODE	FAILURE REASON	REL RANK	FAILURE MECHANISM	POSSIBLE CAUSES	HOW TO ELIMINATE/MIN-IMIZE FAILURE MECHANISMS
A. <u>SOLID TRANSFER LIBRI-CATED BALL BEARINGS</u> with MOS_2 in synthetic ball retainer Loss of Power (Complete or partial)	Excessive losses due to high bearing friction or failure	I	1. Excessive wear of retainer with deb debris fouling ball tracks.	i. Excessive bearing loads.	(a) Balance gear tooth loads. (b) Use spline or coupling for the output drive. (c) Utilize larger bearings. (d) Use MOS_2 coatings on raceways.
				ii. Excessive ball/ball pocket interface loads.	(a) Utilize preload springs or adopt duplex preloaded bearing pair to unify ball velocity and load.
				iii. Excessive temperature.	(a) Use more efficient motor. (b) Use motor concept wherein motor heat flux path bypasses the gearhead. (c) Improve heat flux path from motor mounting.

Table 2 .(cont)

Failure Mecahnism Analysis (cont)

FAILURE MODE	FAILURE REASON	REL RANK	FAILURE MECHANISM	POSSIBLE CAUSES	HOW TO ELIMINATE/MIN-IMIZE FAILURE MECHANISMS
				iv. Skidding of balls during acceleration due to frequent stop/start duty cycles.	(a) Utilize preload to provide ball traction.
				v. Bearing misalignment	(a) Improper housing tolerances or out of roundness at interface with bearing.
B. DRY FILM LUBRICATED BALL BEARINGS i.e., MOS_2 dry film lube on ball races.					
Loss of Power (Complete or partial)	Excessive losses due to bearing friction or failure	II	1. Fouling of ball raceways with dry film lube detritus.	i. Unit did not receive a preliminary run-in followed by a disassembly and cleansing operation.	(a) Improve process control. (b) Specify the exact run-in procedures needed. (c) Have the bearings lubed and run in prior to assembly of shields possibly by the manufacturer, then cleaned prior to final assembly.
C. ALL BALL BEARING TYPES					
Loss of Power (Complete or partial)	Excessive losses due to bearing friction of failure	III	1. False brinnelling (fretting or fretting corrosion) of raceways and balls.	i. Vibration environment without resonance; i.e., motor experiences only input energy.	(a) Utilize duplex preloaded bearing. (b) Operate motor during launch phase (often not feasible).

Failure Mechanism Analysis (cont)

FAILURE MODE	FAILURE REASON	REL RANK	FAILURE MECHANISM	POSSIBLE CAUSES	HOW TO ELIMINATE/MIN-IMIZE FAILURE MECHANISMS
					(c) Use bearing with better static load capacility. (d) Use vibration isolators. (e) Use sleeve bearings (and probably larger motor). (f) Exclude air and oxygen (this eliminates the corrosion aspect).
				ii. Vibration environment with motor resonance.	(a) Redesign motor rotating assembly and bearing stiffnesses to prevent excitation. (b) Operate motor during vibration environment (often not feasible). (c) Utilize duplex preloaded bearing. (d) Use larger capacity bearings. (e) Use vibration isolation. (f) Use sleeve bearings.
				iii. Vibration environment with resonance of supporting structure.	(a) Modify structure to prevent excitation. (b) Any of the solutions (b) to (f) in Item ii above.

Failure Mechanism Analysis (cont)

FAILURE MODE	FAILURE REASON	REL RANK	FAILURE MECHANISM	POSSIBLE CAUSES	HOW TO ELIMINATE/MIN-IMIZE FAILURE MECHANISMS
			2. Brinnelling (plastic deformation of raceways.)	i. Excessive loads from shock environment.	(a) Use larger capacity bearings. (b) Incorporate bearing preloading. (c) Mount motor on shock isolators. (d) Use sleeve bearings.
				ii. Excessive radial load due to incipient gear failure or to overload.	(a) Remedy the gear or overload problem.
				iii. Excessive thrust load due to differential radial interference between housing and bearing (at low temp) restricting relative motion of bearing.	(a) Use housing material with compatible thermal expansion coefficient. (b) Use insert in housing of compatible material to house bearing. (c) Reduce thermal gradient by providing better heat flux path to heat sink.
				iv. Excessive thrust load due to differential linear expansion between housing and rotor combined with inadequate bearing end float allowance.	(a) Use housing material with compatible thermal expansion coefficient. (b) Allow greater bearing and float. (c) Use wet lube instead of solid lube for better heat conduction.
				v. Improper installation (installation loads imposed through balls.)	(a) Implement stricter assembly procedures and quality control measure.

Failure Mechanism Analysis (cont)

FAILURE MODE	FAILURE REASON	REL RANK	FAILURE MECHANISM	POSSIBLE CAUSES	HOW TO ELIMINATE/MINIMIZE FAILURE MECHANISMS
			3. Fretting corrosion at interface between bearing and housing.	i. Excess clearance between bearing and housing reacting to bearing loads (motor operating).	(a) Lessen the clearance between housing and ball race; i.e. use closer tolerance. (b) Use housing material that has similar coefficient of thermal expansion to that of the bearing.
				ii. Resonance of motor structure acting on clearance between bearing and housing (motor operating or not operating).	(a) Redesign motor rotation assembly and bearing stiffnesses to prevent excitation. (b) Use vibration isolation. (c) Use sleeve bearings.
				iii. Resonance of supporting structure acting on clearance between bearing and housing (motor operating or not operating).	(a) Modify structure to prevent excitation. (b) Use vibration isolation. (c) Use sleeve bearings.
			4. Contamination of raceways with gear wear detritus.	i. Unshielded type bearings used.	(a) Utilize shielded bearings.
				ii. Gearing undergoing a failure mode.	(a) Improve gear lubrication. (b) Change to a harder gear material and/or surface treatment of both faces. (c) Eliminate gear misalignment.

Failure Mechanism Analysis (cont)

FAILURE MODE	FAILURE REASON	REL RANK	FAILURE MECHANISM	POSSIBLE CAUSES	HOW TO ELIMINATE/MINIMIZE FAILURE MECHANISMS
				iii. Detritus deposited in bearing as internal pressure is bled to vacuum (pertains to output shaft bearings).	(a) Use vent (with filter) in motor housing to avoid airflow through bearings.
			5. Corrosion of raceways and balls.	i. Unit is subject to humid atmospheres.	(a) Use a sealed motor either hermetically sealed, or sealed with shaft seals or labrinth. (b) Locate motor in sealed section of instrument or spacecraft. (c) Use stainless steel ball bearings.
				ii. Chemical reaction between bearing materials and lubricant.	(a) Change to corrosion inhibiting lubricant. (b) Utilize 440C ball bearings.
				iii. Oil migration loss during storage.	(a) Use barrier film adjacent to lubed area. (b) Use ball retainer that acts as oil resevoirs. (c) Use interference fit bearing shields. (d) Use supplementary oil reservoirs.
			6. Premature failure accompanied by wear.	i. Inadequate lubrication - bearing operates in severe boundary regime.	(a) Use more viscous lubricant and/or better surface finish bearing to achieve partial EH D conditions.

Failure Mechanism Analysis (cont)

FAILURE MODE	FAILURE REASON	REL RANK	FAILURE MECHANISM	POSSIBLE CAUSES	HOW TO ELIMINATE/MINIMIZE FAILURE MECHANISMS
					(b) Use lubricant with better lubricating characteristics. (c) Increase rpm to achieve partial EHD conditions. (d) Chemically contaminated balls &/or raceways causing non-wettable surfaces.
				ii. Inadequate lubrication due to evaporation loss.	(a) Use sealed motor either hermetically sealed or sealed with shaft seals or labrinth. (b) Utilize oil or grease with lower evaporation rate. (c) Locate motor in sealed section of instrument or spacecraft. (d) Use ball retainers that can act as oil reservoir and add supplementary reservoir. (e) Reduce extent of preflight testing. (f) More frequent lubrication refurbishments.
				iii. Inadequate lubrication due to migration loss.	(a) Use barrier films adjacent to lubed area. (b) Use ball retainers that act as oil reservoirs.

Failure Mechanism Analysis (cont)

FAILURE MODE	FAILURE REASON	REL RANK	FAILURE MECHANISM	POSSIBLE CAUSES	HOW TO ELIMINATE/MINIMIZE FAILURE MECHANISMS
					(c) Use interference fit bearing shields. (d) Use supplementary oil reservoir.
				iv. Excessive temperature causing degradation of wet lube (polymerization) followed by severe wear.	(a) Use more efficient motor. (b) Use motor concept wherein heat flux path bypasses gearhead. (c) Improve heat flux path from motor mounting. (d) Use lube with higher temperature capability. (e) Reduce normal load or preload on bearings. (f) Change from grease to oil lubrication.
			7. High oil viscosity.	i. Actual operating temperature is lower than anticipated.	(a) Change to an oil or grease with lower temperature/viscosity coefficient and less viscous at the low temperature. (b) Change to dry film lube. (c) Use more powerful motor. (d) Use heater to avoid the low temperature condition.

Failure Mechanism Analysis (cont)

FAILURE MODE	FAILURE REASON	REL RANK	FAILURE MECHANISM	POSSIBLE CAUSES	HOW TO ELIMINATE/MINIMIZE FAILURE MECHANISMS
				ii. Oxidation of lubricant.	(a) Use an oxidation inhibiter. (b) Use a more powerful motor.
				iii. Lubricant properties deviate from specification.	(a) Impose stricter quality control procedures.
			8. Contamination of raceways with brush debris.	i. Vibration or maneuver forces disturbs debris causing it to enter bearing.	(a) Use shielded or sealed bearing. (b) Use flange/slinger on shaft adjacent to bearing.
				ii. Brush debris deposited in bearing as internal pressure is bled to vacuum (pertains to output shaft bearings).	(a) Use vent (with filter) in motor housing to avoid airflow through bearings. (b) Use contact type shaft seal (and inert gas pressurization). (c) Hermetically seal. (d) Avoid use of brush type motor.

Failure Mechanism Analysis (cont)

FAILURE MODE	FAILURE REASON	REL RANK	FAILURE MECHANISM	POSSIBLE CAUSES	HOW TO ELIMINATE/MINIMIZE FAILURE MECAHNISMS
			9. Ball retainer wear, combined with race contamination leading to fatigue and rupture of retainer	i. Loss of lubricant	(a) Change from a ball riding retainer to a land riding retainer, oil impregnated. (b) Use supplementary oil reservoirs.
				ii. Excessive ball/ball pocket interface forces due to excessive bearing loads.	(a) Balance gear tooth-load. (b) Reduce bearing pre-load.
				iii. Excessive ball/ball pocket interface forces due to widely varying ball velocities and loads.	(a) Add preload - but check hertz stress and fatigue life.
				iv. Excessive friction between land riding ball retainer and land (particularly at extremes of temperature).	(a) Increase clearance between retainer and raceway. (b) Use lower viscosity oil, or change from grease to oil. (c) Use a more dimensionally stable retainer material.

Failure Mechanism Analysis (cont)

FAILURE MODE	FAILURE REASON	REL RANK	FAILURE MECHANISM	POSSIBLE CAUSES	HOW TO ELIMINATE/MINIMIZE FAILURE MECHANISMS
				v. Skidding at ball during acceleration (particularly in frequent start/stop duty cycles).	(a) Utilize preload to provide ball traction
				vi. Bearing misalignment due to improper housing tolerances.	(a) Remedy faulty Q.C. procedures.
				vii. Bearing misalignment due to out-of-roundness at housing bore.	(a) Improve machining expertise. (b) Use material with better dimensional stability.
				viii. Improper fit of shaft onto inner race of bearing with consequent creep, scoring and cracking.	(a) Use correct tolerance and size code.
				ix. Shaft bending (due to gear load).	(a) Use larger shaft or shorter bearing span. (b) Balance gear lead.
			10. Spalling of raceways (and sometimes balls); classical subsurface fatigue initiated by subsurface imperfections.	i. Hertz stresses much higher than anticipated due to improper choice of bearing.	(a) Retrofit larger capacity bearings.
				ii. Inadequate determination of statistical reliability - failed bearing is a low life outcome of the Weibull distribution.	(a) Retrofit bearings, adequately derated for load, to provide the required life at 99.9% to 100% reliability.

Failure Mechanism Analysis (cont)

FAILURE MODE	FAILURE REASON	REL RANK	FAILURE MECHANISM	POSSIBLE CAUSES	HOW TO ELIMINATE/MINIMIZE FAILURE MECHANISMS
				iii. Excessive hertz stresses due to large race radius.	(a) Utilize bearing with smaller raceway radius (but higher friction).
				iv. Excessive portion of fatigue life expended during vibration environment.	(a) See Item iii above.
				v. Bearing not operating in favorable lubrication regime. Hertz stresses are therefore higher than otherwise.	(a) Use more viscous lube, higher speed, better surface finishes and precision grade bearings.
				vi. Inferior metal (high inclusion density)	(a) Improve quality of steel. (b) Use selective portion heat; i.e. impose stricter quality control.
			11. Spalling of raceways (and sometimes balls) due to surface initiated fatigue.	i. Non-metallic inclusions coincident with surface of ball race, causing pits (micro-spalling).	(a) Enforce quality control standards on steel supplier, or improve quality control standards. (b) Select a better quality steel. (c) Impose a more rigid finished part inspection routine.
				ii. Wear debris from asperity interaction imbedding or causing dents in ball path, from which spalling nucleates.	(a) See Item C 9 above.

FAILURE MODE	FAILURE REASON	REL RANK	FAILURE MECHANISM	POSSIBLE CAUSES	HOW TO ELIMINATE/MINIMIZE FAILURE MECHANISMS
				iii. Lubricant contaminant causing dents and stress concentrations in ball path which initiates spalling.	(a) Use stricter quality control on lubricant. (b) Improve cleanliness of assembly site. (c) Prevent ingress of foreign matter through labrinth seal by installing filter/ breather in motor housing.
				iv. Grinding and housing imperfections - nicks and furrows, causing stress concentrations.	(a) Use stricter quality control measure. (b) Demand high quality (more expensive) bearings.
			12. Contaminated lubricant.	i. Inadequate cleanliness during fabrication and assembly.	(a) Impose more rigerous cleanliness provisions on motor and bearing manufacturers.
				ii. Inadequate lubricant cleanliness.	(a) Impose more rigerous cleanliness provisions on lubricant supplier, and/or; (b) Microscopically select lubricant for cleanliness for each bearing.
			13. Encroachment of ball track on race land.	i. High thrust on ordinary (conrad) type bearing (due possibly to differential thermal expansion between casing and rotor).	(a) Use precision grade bearing. (b) Use angular contact type bearing. (c) Allow more bearing end float.

Failure Mechanism Analysis (cont)

FAILURE MODE	FAILURE REASON	REL RANK	FAILURE MECHANISM	POSSIBLE CAUSES	HOW TO ELIMINATE/MINIMIZE FAILURE MECHANISMS
			13. Encroachment of ball track on race land (cont)	ii. High thrust on ordinary, conrad type bearings (due possibly to differential radial interferences between housing and bearing - at low temperature - restricting relative motion of bearing).	(a) Use bearing housing material with compatible thermal coefficient of expansion. (b) Perform worse case analysis and correct errors in dimensioning.
			14. Retainer instability.	i. Inadequate ball/ball pocket lubrication.	(a) Utilize porous oil impregnated ball separator.
				ii. Inadequate clearance or lubrication at separator/race-land interface.	(a) Utilize porous oil impregnated ball separator. (b) Adjust separator/race land clearance to provide hydrodynamic operation. (c) Use oil with more suitable viscosity range. (d) Change from outer race to inner race riding separator, or vice versa. (e) Use preloading to unify ball velocity and ball/ball pocket forces.

Failure Mechanism Analysis (cont)

FAILURE MODE	FAILURE REASON	REL RANK	FAILURE MECHANISM	POSSIBLE CAUSES	HOW TO ELIMINATE/MINIMIZE FAILURE MECHANSIMS
D. <u>PLAIN BEARINGS</u> (as used for planet gears of some gearheads). Loss of Power (complete or partial)	Excessive losses due to bearing friction	IV	1. Excessive wear or galling of bearing interfaces.	i. Degraded dry film lubrication.	(a) Use a more tenacious dry film lube. (b) Change to a wet lube (with output shaft seals). (c) Utilize ball bearings in planet gears.
				ii. Loss of wet lubricant	(a) Use sealed motor either hermetically sealed with shaft seals or labrinth. (b) Locate motor in sealed environment. (c) Use grease lubricant instead of oil. (d) Use oil or grease with lower evaporation rate.
			2. High lubricant viscosity.	i. Actual operating environment is colder than anticipated.	(a) Use oil or grease with lower temperature/viscosity coefficient and less viscous at low temperature. (b) Change to dry film lube. (c) Use a heater to avoid the low temperature condition.

Design

Since bearings and lubrication are the acknowledged and proven life limiting elements of motors, emphasis has been placed on the examination of bearing fatigue life and reliability and the types of lubrication systems which enhance long life. Design features are explained by reference to illustrations of successful schemes in use at the present time.

Emphasis has also been placed on determining practical methods of combatting the severe vibrational conditions that will be experienced in applications on Shuttle type of programs. In addition, methods of bearing installation which overcome the problem of differential thermal expansion between motor housing and rotating assembly are reviewed.

The review of bearings and lubrications systems will be limited to ball bearings as utilized in motors for space applications where temperatures extremes do not exceed -200°F to 300°F.

While the purely electrical elements of motor construction have not exhibited spaceflight problems to date, it is conceivable that the more severe environments of space Shuttle type programs could have drastic effect on potted windings, hermetic glass sealed terminals and electrical conductors. Except to suggest that current state-of-the-art motors should be subjected to investigation under the increased severity of environment, no attempt will be made in this study to provide solutions to these potential problem areas.

a. Bearings and Lubrication Systems - Wear must be virtually eliminated for bearings to achieve their ultimate life capability, i.e., their fatigue life. Fluid lubrication (with nominal film thicknesses) offers this opportunity. In fact, fatigue life is predicated on this. Hence for long life, every reasonable effort must be pursued to facilitate the successful incorporation of fluid lubrication.

Fluid lubrication also offers the opportunity of operating in the partial or full elastrohydrodynamic regimes of lubrication where substantial improvements in life are achievable. EHD theory has developed to a level of usefulness. The latest ASME recommendation in this regard will be discussed and related to aerospace requirements later in this section.

Lubrication selection is almost always a compromise. Not only must the selected lubricant provide adequate lubricity and visco pressure characteristics over a specified temperature range, but its oxidation and thermal stability will assume more or less importance according to the type of environment and duty. For instance, in certain motor applications, a relatively small increase in torque due to lubricant oxidation or polymerization could constitute failure. Also of paramount importance is evaporation rate and vapor pressure. Aggravitating the overall situation, is the fact that, whereas high viscosities can place a slow speed bearing in a favorable lubrication posture, friction will increase (according to the 0.66 power of viscosity) thus demanding more power from the motor and introducing still further compromises.

Obviously, a lubricant with a favorable viscosity-temperature coefficient is preferable for extreme variation of operating temperature. However, final selection must take into account the amount of running at any particular temperature such that viscosity range will avoid boundary conditions for the maximum amount of operational life.

When appreciable periods of boundary operation are unavoidable, special emphasis should be placed on selecting a lubricant with good boundry lubrication characteristics. Unfortunately, the silicone lubricants which have excellant viscosity index rating are notoriously poor boundry lubricants.

Operating temperature levels not only dictate the type of lubricant to be used, but associated structural expansions and contractions must be accommodated. Methods of overcoming these problems are reviewed under the caption of "Bearing Selection and Installation." These techniques pertain regardless of whether solid or fluid lubrication is used.

Heat generation in the bearing as a result of ball skidding or sliding and retainer instability must also be considered. While preloading tends to unify ball loading and alleviate ball skidding and sliding and the ball/ball retainer interface problems, the additional load increases torque, friction and heating. In the case of induction motors, the fact that rotor heat must be conducted through the ball contacts into the housing further complicates the issue.

Amelioration of this situation can be achieved if the cruciality of temperature to long life lubrication is recognised and action is taken during the conceptual stages of spacecraft design to locate motorized devices as beneficially as possible. For comprehensive information on the wide variety of fluid lubricants available, reference should be made to Part B of the *Lubrication Handbook for Use in the Space Industry* available from:

D. E. Lyons
A&TS-PR-M
Office of Procurement & Contracts NASA
Marshall Space Flight Center
Huntsville, Alabama, 35812.

1) Fatigue Life Adjustment Factors - The basic dynamic capacity of bearings, based on fatigue life, was established by the Anti Friction Bearing Manufacturers Association (AFBMA) in 1949. It still represents the basic datum point from which actual load/life capability of a particular bearing arrangement may be predicted or extrapolated. It assumes that the bearing will fail from no other reason than stress fatigue and that the following conditions will prevail:

1) Proper installation; i.e., no misalignment, etc;

2) Favorable fluid lubrication; now regarded as partial EHD;

3) Through hardened steel races of 58 Rc hardness;

4) Nonflexible bearing rings;

5) Zero clearance;

6) Inner race rotation;

7) No contamination.

Fatigue was assumed to result from subsurface shear stress reversals as the balls passed through their loading cycle. It is precipitated by a subsurface imperfection in the races, resulting in a classical spall failure. Since the incidence and dispersion of imperfections is random in nature, the establishment of life values was necessarily based on statistical evaluation of multiple test results. (This effort represents one of the earliest and now classical examples of Weibull statistical analysis.)

Since 1949 numerous advances associated with melting, metallurgy and manufacturing techniques have improved the life potential of bearings considerably. Also, considerable insight has been gained into the mechanism of rolling element lubrication, resulting in the technology of elastohydrodynamics. This takes into account the elastic properties of the rolling elements in conjunction with changes in viscosity of the lubricant due to the pressure at the rolling element interface.

The manner in which this new technology affects bearing life capability has recently been investigated by the Rolling Elements Committee of the Lubricants Division of the ASME. They have published a very useful design guide entitled: *Life Adjustment Factors for Ball and Roller Bearings*, dated September 1971. This design guide, formulated by foremost specialists from the bearing industry and research institutions, is based on the results of world-wide investigation. Sixty-seven research programs and technical papers are covered and referenced. It identifies five multiplicative factors by which to modify the basic relationship:

$$L_{10} = \left(\frac{C}{P}\right)^n$$

such that

$$L_A = (D)\ (E)\ (F)\ (G)\ (H) \left(\frac{C}{P}\right)^n$$

where:

C = Basic load rating;

P = Static equivalent load;

n = Load life exponent; 3 for ball bearings, 10/3 for roller bearings;

D = Life adjustment factor for materials technology improvement;

E = Life adjustment factor concerning metallurgical processing improvements;

F = Life adjustment factor dependent on lubrication regime;

G = Speed effect factor;

L_A = Expected bearing life.

It is important to observe (in the words of the design guide) that the above factors merely; "extend the engineering approximations which are illustrated in most bearing manufacturers catalogs and provides information that will be of most use to the engineer."

Usually the above factors result in a life improvement; however, in the case of factor "D", materials other than 52100 result in a life reduction. It will also be observed that 440C stainless steel (which is often used in spacecraft applications to avoid possible corrosion) is not listed. In view of the considerable difference of opinions concerning the relative life capability of 52100 and 440C, it is suggested that a value of D = 1.0 be used for 440C, pending authorative data. However, the same processing factor E_1, of 3.0, should be used for 440C, providing that it is consumable electrode vacuum melted.

In regard to the lubrication factor F, it will be noted that many of the customary aerospace lubricant, including silicones, fluorosilicones, synthetic hydrocarbons and perfluoroalkypolyether are not referenced; hence, the lubricant manufacturer must be consulted for viscosity/pressure characteristics and the $(\mu a)^{0.7}$ value computed independently of the design guide.

An important feature which is not discussed in the design guide is the problem of lubrication starvation. This can occur in non-flooded bearings due to the inability of the oil film to replenish itself between passage of one ball and the next. This phenomena could very readily pertain to spacecraft bearings since they usually have to operate with minimal amounts of lubricant. Hence, every effort should be made to operate with a high Λ factor. The use of optimum surface finishes on the rolling elements is one means towards this end; however, the validity of high Λ factors achieved by ultra fine surfaces is still unproven. Apparently, wavyness and accuracy of rolling element geometry also enters into this consideration.

The speed factor which takes centrifugal force into account will not be applicable in the vast majority of motor bearing application because their DN values will seldom exceed 0.4×10^6.

Having applied all these factors to the catalogue $\left(\frac{C}{P}\right)^3$ value, we arrive at an adjusted L_{10} life; but we have not improved on the nominal 90% reliability. This is unacceptable for aerospace applications. Figure 12 provides a reliability adjustment factor applicable to either life or to load rating $\left(\frac{C}{P}\right)^3$ function.

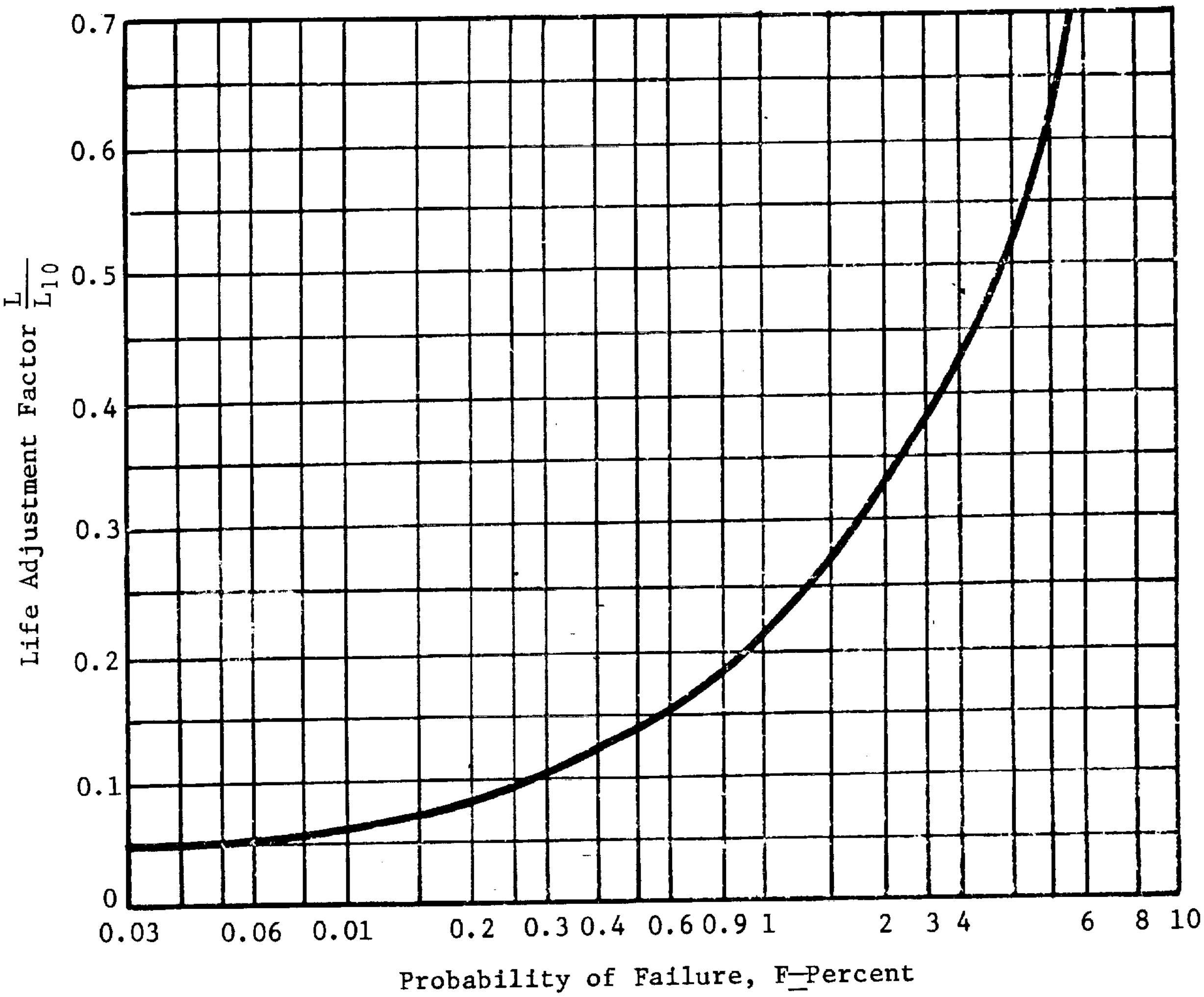


Figure 12 Life Adjustment Factor for Reliability

Courtesy of SKF Industries
King of Prussia, Pennsylvania

One-hundred percent reliability may be achieved by factoring the basic dynamic load rating "C" by $(0.05)^{1/3}$; i.e., C is reduced by 63%. The effect of this no failure derating factor is to reduce contact stresses to a level below the endurance limit. Alternatively, at maximum load rating, $L_{nF} = 0.05\ L_{10}$, ninety-nine percent reliability is achieved with a factor of 0.2, equivalent to a load rating reduction of 42%.

The foregoing life and load rating and derating considerations pertain only to fluid lubricated bearings and assumes that there is no failure mode due to wear. It is of little benefit to the sizing of dry film lubricated bearings; however, such bearings are less likely to be selected for long-life applications. When they are, the guess and test method of selection is the only approach using the limited amount of published test data as a guide. This topic is reviewed in Section C.2.a.4.

It is reemphasized that the foregoing life/load rating system and environmental factors provide "engineering approximations" and that: "very often these environmental factors, considering varying applications, bearing configuration, and bearing rating, become exceedingly complex so that problem solution in reasonable time requires high speed computers and more information than is readily available to most engineers."

2) Bearing Vibration Problems - Vibration environments have always been a hazard to rolling contact bearings particularly if resonant modes were experienced. With the more severe vibration environments predicted for Shuttle type operations with input levels reaching 73 gs rms or more, the bearing problem assumes major proportions. However, not only are the g levels very high, but the duration of exposure is long. Instead of the usual few minutes, the duration may amount to 5 hours in each axis. In view of these extreme vibration requirements, it is desirable to design the motor and its support structure such that resonant frequencies are either sufficiently remote or will occur at low energy levels of the input spectrum, such that system response (amplification) does not impose intolerable force levels.

As far as the rotating assembly is concerned, two stiffness values impose different resonant modes; actual resonance being a resultant of both modes. The inherent structural stiffness of the motor shaft determines one mode, the other by the stiffness of the bearings. Shaft stiffness is a function of its length and diameter;

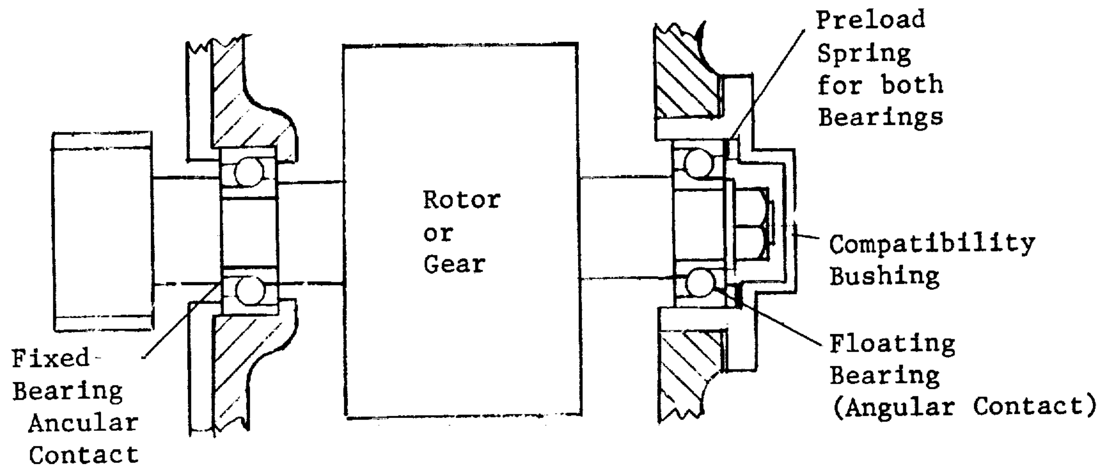
thus the pancake type motor may have advantages in that its shorter, larger diameter shaft will tend to raise the resonant frequency when viewed as a beam. This can be adjusted by varying the bearing preload, race geometry and ball compliment.

In the axial direction, bearing stiffness is the primary determinant of resonant frequency and represents a simple single degree of freedom system. In the lateral direction, a two degree of freedom system is involved because of the two bearings. Also two different resonant frequencies are involved unless both bearings are of equal stiffness and rotor is symmetrical. This is where a prime difficulty lies since preloaded (angular contact) bearings need to be incorporated at each end of the motor shaft in such manner that ambient and operational temperature changes will have insignificant affect on preloading. Furthermore, it is preferable to utilize preloaded bearing pairs since these provide stiffnesses in the order of millions of pounds/inch; but it is even more difficult to incorporate these at both ends of a shaft.

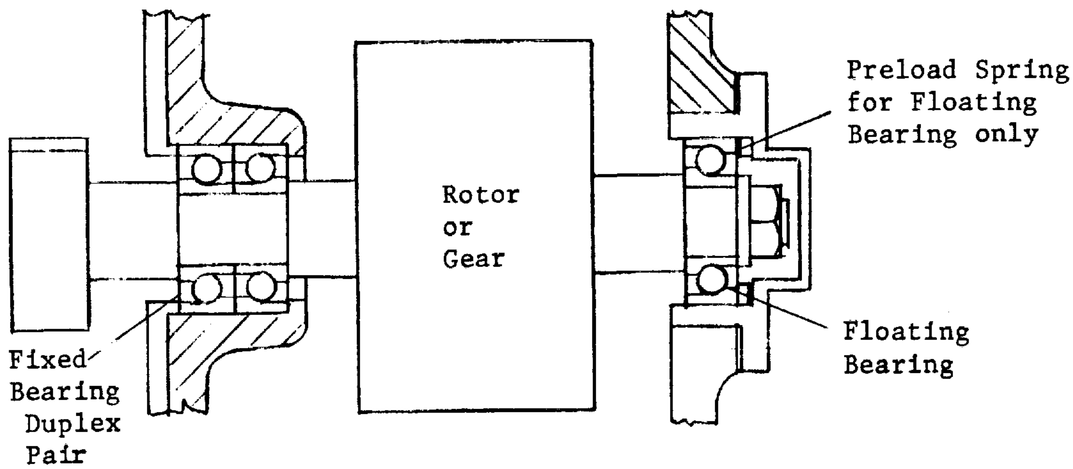
The arrangements of Fig. 13b and 13c are probably the best practical compromises using readily available bearings. However, while the high stiffness of the duplex bearing is beneficially utilized to achieve a high resonant frequency in the axial direction, it doesn't have the same benefit in the transverse direction since the floating bearing will lower the resonant frequency in this direction. Ideally, the floating bearing should have a radial stiffness equal to that of the anchored bearing.

Figure 14 may offer a more optimized solution. It uses a radially preloaded roller or ball bearing at the floating end of the rotor. This also offers the advantage that the outer race of the "floating" bearing no longer needs to float. The rolling elements, although preloaded, can accommodate this feature. It will be noted in the case of the floating ball bearing that the outer race contains no raceways thereby leaving the balls free to travel axially relative to the outer race. The inner raceways would restrain the balls axially. Since the ungrooved outer race is liable to introduce high contact stresses, two rows of balls may prove advantageous as illustrated.

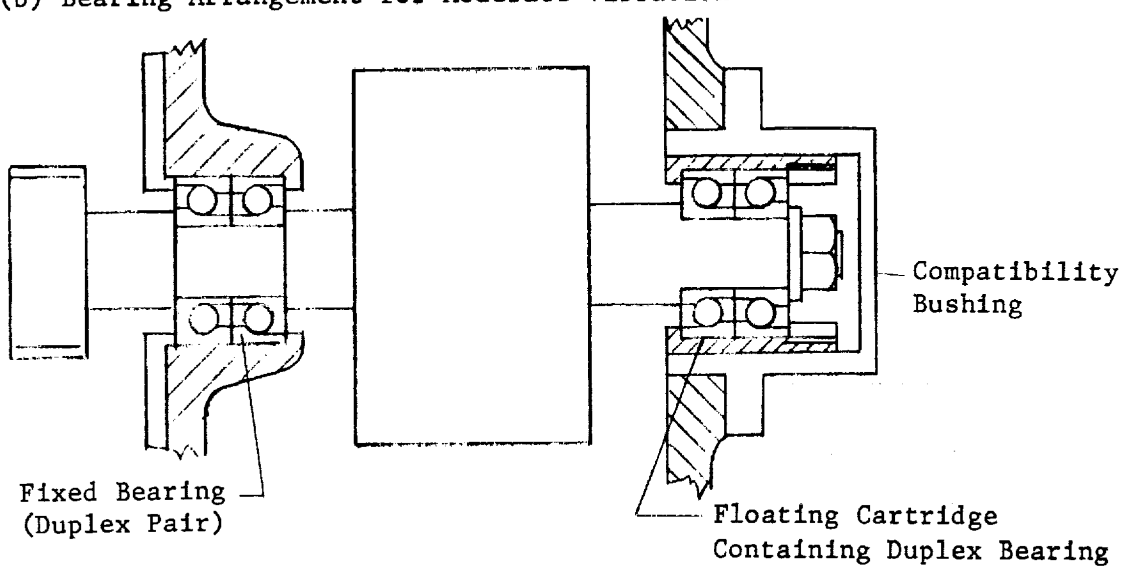
Existing methods of radial preloading of bearings utilize a tapered bore in the inner race which engages with a mating taper on the shaft. A nut is then used to force the inner race along the taper. The resulting expansion creates a radial preload. The disadvantage of this method is that it relegates the responsibility of acquiring the correct preload to the motor manufacturer.



(a) Bearing Arrangements for Mild Vibtation

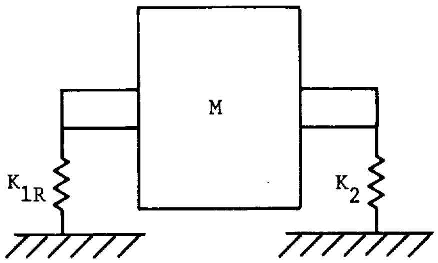


(b) Bearing Arrangement for Moderate Vibration



(c) Bearing Arrangement for Severe Vibration

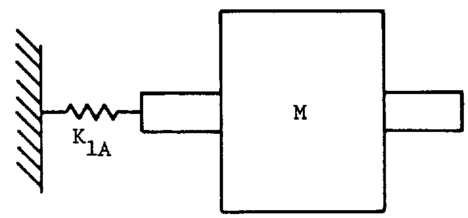
Figure 13 Bearing Arrangements for Vibration



Two Degrees of Freedom in Radial Mode

K_{1R} = Radial Stiffness of Duplex Pair

K_2 = Radial Stiffness of Roller Bearing



Single Degree of Freedom in Axial Mode
K_{1A} = Axial Stiffness of Duplex Bearing

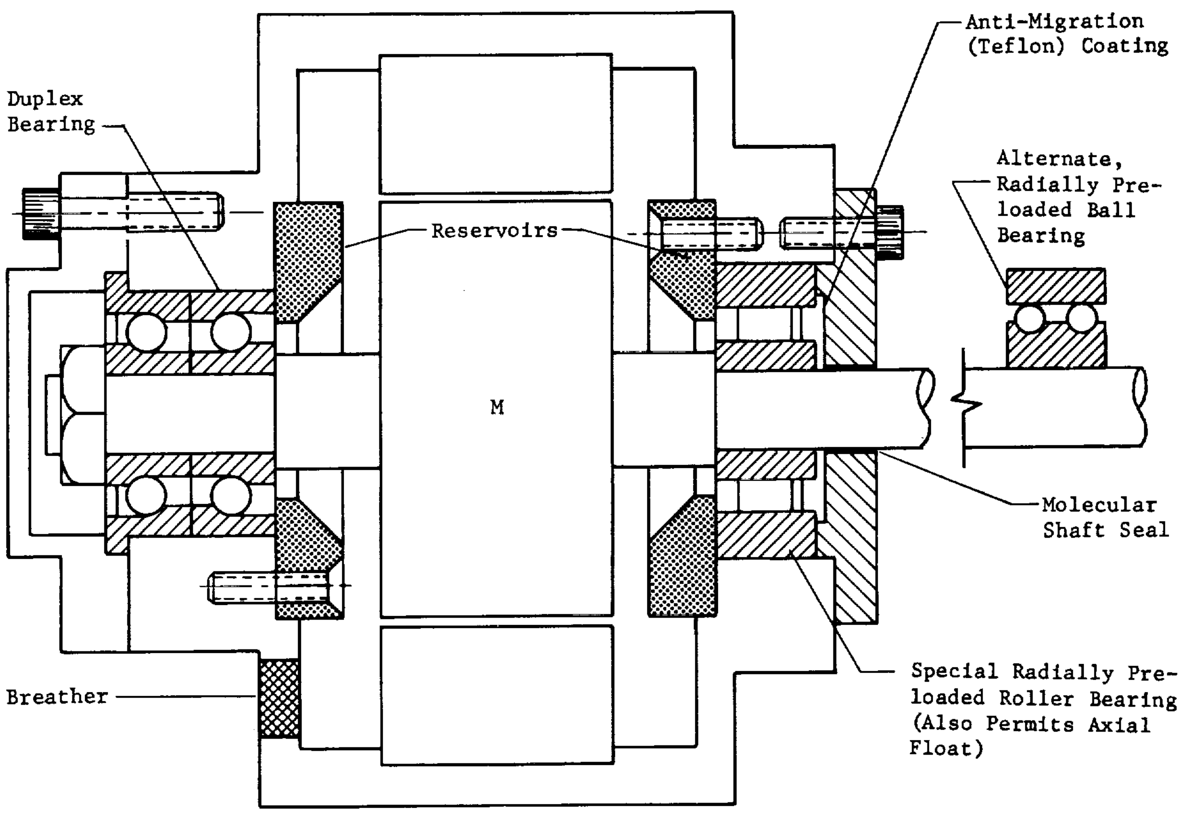


Figure 14 Bearing Preloading Concept

The scheme of Fig. 14 avoids this problem and would permit the bearing manufacturer to control the preload by precision control of dimensions. The outer race would have to be assembled over the rollers or balls at an elevated temperature. Selection of the appropriate ball or roller diameter is the obvious means of determining the extent of preload. Cross sections of races should be substantial to avoid dimensional change and resulting change in preload and stiffness caused by interface fits in housing or on shaft. ABEC 9 tolerances would be required to render this practical. (The writer is unaware of any existing application of these concepts. Because of the importance of minimizing vibration failures of bearings, especially in view of the seriousness of the problem on Shuttle, it is submitted that these concepts should be explored by means of a separate study contract.)

The preloading necessary to achieve a specific bearing stiffness may well exceed the loads imposed as a result of motor operation. Hence, the bearing selected must not only provide the required stiffness, but it must provide the required life at 100% reliability at the load which provides this stiffness (plus operational load).

By avoiding resonance in this manner, system response is minimized, i.e., the motor will be subject to minimum amplification of input g levels. However, this could still involve several hundred gs rms during certain flight regimes and motor location in the spacecraft. Hence a further demand on the bearing is to resist these vibrationary forces. In some instances, the motor may be operating; but in many other instances, it will be stationary depending on type of motor application.

If stationary, the static load capability and the fatigue rating of the bearing must now be assessed. For instance, if input vibration energy can be considered to concentrate at an average frequency of 1000 Hz, then the total number of stress cycles on the bearings for the two transverse axes is:

10 hr x 3600 x 1000 = 36 x 10^6 cycles;

which, dependent on stress could well initiate surface fatigue.

Bearings must, therefore, be sized to provide--

1) the necessary stiffness to avoid resonance and severe amplification;

2) adequate fatigue life under normal operating conditions;

3) adequate capability (under vibration, fretting corrosion conditions) to withstand millions of static load cycles without surface degradation sufficiently severe to effect normal life expectancy.

It is to be noted that the derating procedure recommended in Section C.2.a.1 to use a larger bearing than would otherwise be selected, also provides a higher static load capability (which is not derated). Thus it may meet the requirements of 3) above without resorting to an even larger bearing.

Vibration loading of a stationary ball contact involves plastic flow combined with a fretting corrosion type of wear. The latter is due to the fact that there is no rolling contact to develop a film. In the case of a rotating ball contact operating in the EHD regime, a lubricant film cushions the effect--film thickness being relatively immune to load. Stationary ball contacts have been investigated (Ref 2 and 3) at accelerated loading rates of 28,000 Hz and involving Hertz stress levels to 750,000 psi.

In these experiments, false brinelling (fretting corrosion) left indentations up to 50 microinches deep; whenever fatigue and spalling occurred, identations were twice this depth. Both values exceed the criteria for static load rating, i.e., that deflection should not exceed 0.0001 of the ball diameter. However, Hertz stress levels were very high. It is submitted that static Hertz stress levels be limited to the conventional figure of 300,000 psi; then the effect of fretting corrosion due to vibration on normal life should not be severe.

Any of these preloading provisions will have a substantial effect on bearing torque which will have to be accommodated by the motor. Hence, severe vibration environment results in a larger, heavier motor than would otherwise be necessary.

If all these measures fail to alleviate severe amplification of input levels, vibration isolation can be resorted to; but this is often impractical. The only remaining solution is to design the

bearings to accept the full impact of an unfavorable system response (i.e., high amplification factors). The design must then accept the complications and expense which accompanies a limited bearing life; i.e., the maintenance of a replacement and refurbishing service. If this latter approach is adopted, then static load rating will probably be the major criteria governing bearing selection.

The resonant frequency of the motor housing must not be overlooked. It will also need to be tailored to the situation. In this case it may be advisable to adopt a low natural frequency; however, this must be decided in conjunction with a knowledge of the input spectrum. It may well be that the weight increase necessary to curtail vibration amplification will be prohibitive, in which case a short bearing life may again have to be accommodated.

As mentioned previously in regard to temperature extremes, every effort should be made during the conceptual stages of spacecraft design, to locate motorized devices in environmentally advantageous localities.

3) Fluid Lubrication Systems - Even when using the lowest evaporation rate lubricants, it is usually beneficial to isolate the motor from space vacuum. For tape recorder and sealed scientific instrument motors, and those in habitation areas, the problem doesn't exist. But for many motors, the only solution is to seal the motor itself. Several different methods of sealing will be examined, using proven examples, as a means of description.

a) Shaft Seals - In the case of gearhead motors where considerable torque is developed at the output shaft and speed is relatively low, shaft seals have proved very successful. However, this is limited to temperature environments in which elastomeric seals can survive. With this arrangement, the motor is pressurized with an inert gas.

The Mariner Gimbal actuator of Fig. 6 is a very successful example. During its 229 day journey to Mars, it lost only 10% of its initial 30 psi pressurization. With the leakage rate diminishing exponentially, it should remain adequately pressurized for many decades.

Since G300 silicone grease with an extremely low evaporation rate was used, there may be some argument as to whether seals and pressurization were necessary for retension of lubricant and suppression of evaporation rate. But they did serve to exclude humid atmospheres and prevent corrosion. Hence, this concept should be of particular benefit to Shuttle type operations where environments will alternate from space-vacuum to humid air.

The linear motion output rod of this actuator is sealed with two viton 'o' ring seals for which very exacting grooves were found to be essential. The unit was designed for 50,000 hr life (60,000 90% full amplitude cycles) at motor speeds up to 700 rpm. Superior 384 brush material was used. Wear during an accelerated 100,000 hr test amounted to 6%.

b) Hermetic Sealing - The advantages of hermetic sealing is delineated in the Introduction together with figures exemplifying several types, i.e.:

The Planocentric hermetic sealing arrangements are shown in Fig. 2 and 4. The main purpose of these particular sealing arrangements was to prevent contamination, not to prevent lubricant evaporation. In fact, the example of Fig. 4 used solid lubrication throughout.

Nevertheless the principle could be utilized to curtail evaporation loss, but it does involve penalties. The bellows introduces inefficiency and is also a fatigue risk; it would need to be over designed for assurance of long life. Except for low duty cycle applications, a brushless type motor would be essential.

Wabble Mechanism Sealing is another hermetic sealing method applicable to low speed output shafts. It is illustrated in Fig. 15. This is a Wabble mechanism and comprises an orbiting bevel gear (probably should be termed a swash gear) and a bellows which accommodates the resulting swash plate type of motion and seals at the same time. Original design life was 10,000 hr continuous operation for OGO, but it has already acquired 30,000 hr of operation without failure. It has seen 2 years of use on Nimbus VI and will be used on future Nimbi and ERTS. In this case, the concept was specifically evolved to enhance long life lubrication. Internal lubricants are Bray Oil's NPT 4 and KK949B. The slow speed bevel gear mesh is exposed to space vacuum and must be dry film lubricated. The welded metal bellows which must react to the output torque is sufficiently large and over-rated that it constitutes no fatigue hazard.

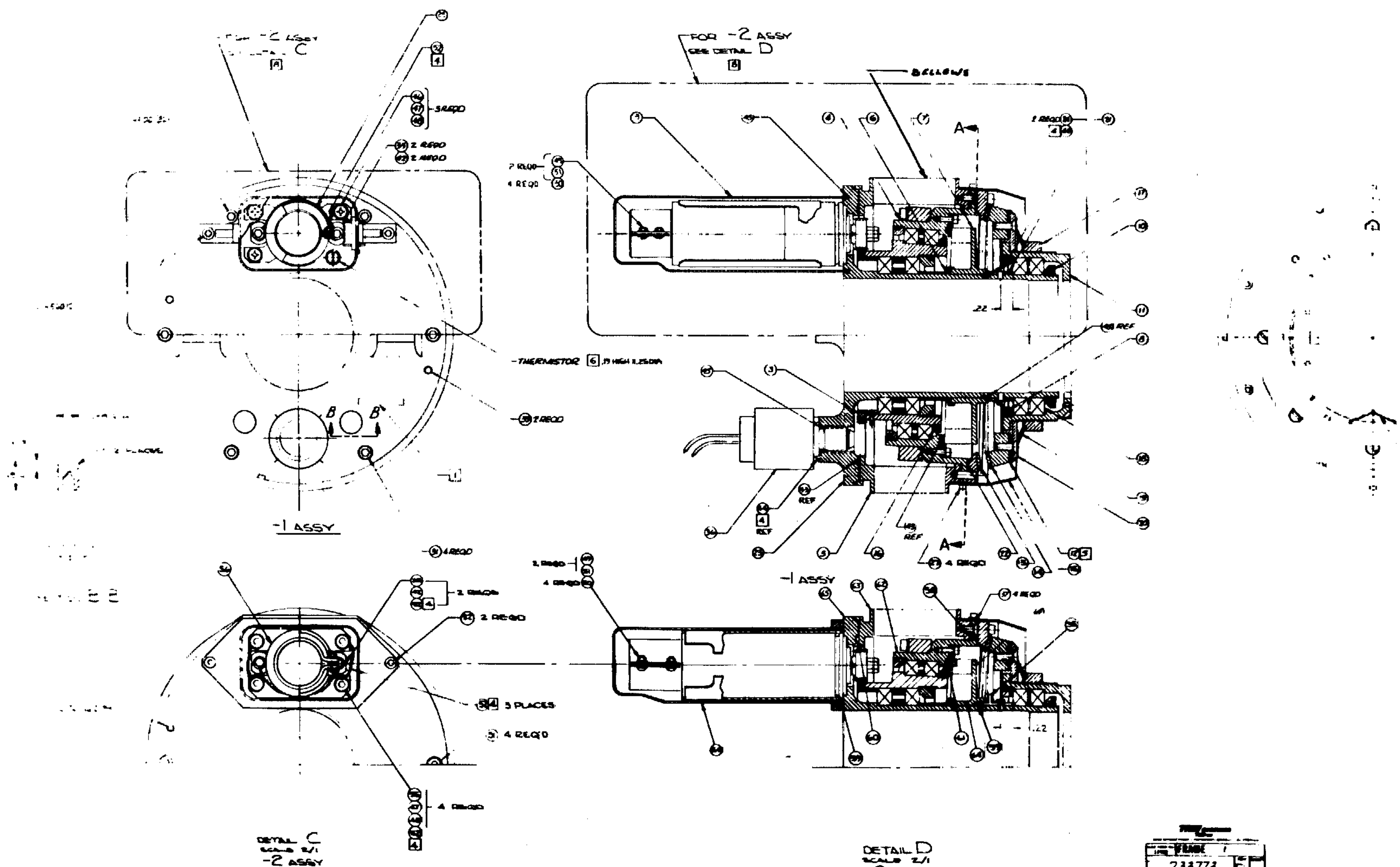


Figure 15 Wabble Mechanism

Courtesy: TRW

The magnetic coupling, as explained earlier, is also readily adaptable to provide hermitic sealing but it is limited to low torques and must be applied to the motor shaft. It is ideal for motor drives for pumps and fans. But it loses much of its advantage when a gearbox must be interposed since it cannot conserve the gear lubricant.

c. *Molecular Seals and Reservoirs* - Ungeared motors usually lack sufficient torque to overcome the drag load of contact type or hermetic type shaft seals. While this definitely applies to the conventional high speed type of motor, it very often applies to the slow speed pancake type of motor as well. Low speed motors often operate in servo-systems which cannot tolerate high levels of seal friction.

In these circumstances, the "molecular" seal has proved satisfactory. It consists of a close clearance fit between output shaft and housing. Clearances have varied according to application from 0.0003 to 0.004 in., depending very largely on the precision grade of bearing utilized.

Such clearances are really controlled leakage paths operating at the molecular flow level, hence the path does not need to be labrinthine. In fact, the normal microscopic surface irregularities act as a flow retardant.

The vapor pressure trapped in the motor housing will depend on the type of oil and the ambient temperature. It will usually reside between 10^{-3} and 10^{-6} torr, resulting in substantial reduction in evaporation rate compared to that at full space vacuum. Obviously, for long life, some means of replenishing the leakage loss must be embodied into the motor without prejudicing the active lubricating films within the motor. Several methods have been utilized with apparent success, i.e.:

1) Oil impregnated, porous reservoirs, and;

2) Oil impregnated ball retainers.

Some reservoir systems are intended to function by evaporation and condensation, others by creepage and some rely on both processes. In neither case is the exact mechanism very well understood.

1) *Reservoirs* - Reservoirs have been sized using the following formula for weight loss rate (derived from Ref 4, 5 and 6.)

$$W = 0.0583\ P \left(\frac{M}{P}\right)^{\frac{1}{2}} f\ A \text{ grams/second.}$$

Where:

P = vapor pressure of the oil in torr

M = molecular weight of vapor

T = temperature of gas in degrees Kelvin

A = area of flow path in cm^2

f = $\pi a/1$, a/1 should exceed 16

a = width of flow path

1 = length of flow path.

Care must be exercised to utilize molecular weights truly representative of the vapor fraction involved. Excess capacity should be provided since it is still conjecture as to whether molecular seals may be subject to liquid creep of condensed lubricant vapors (Ref 7).

Despite the unknowns regarding these reservoir and sealing systems, they seem to have imparted a large measure of success. For instance, the arrangement of Fig. 16 on ATS III operated satisfactorily for four years prior to shut down due to mechanical/electronic difficulties. It used Apiezon C oil with a long chain polar molecular additive for superior adhesion (BBRC's Vackote). Although large molecular seal clearances of 0.004 in. were used, lubricant loss rate was calculated at 0.1 gram/year (giving a total life of 240 years) using the same basic formula as given above.

It will be observed in Fig. 16 that the reservoir surrounds the motor stator which is the hottest portion of the assembly. This promotes evaporation and creates an internal pressure which will retard evaporation from cooler areas. The loss through the seal will be replenished primarily from the reservoir and not from active lubricant films. In fact, these films may be augmented by condensation. Unfortunately, this augmentation may be of a lighter fraction with consequent dilution and decrease of viscosity.

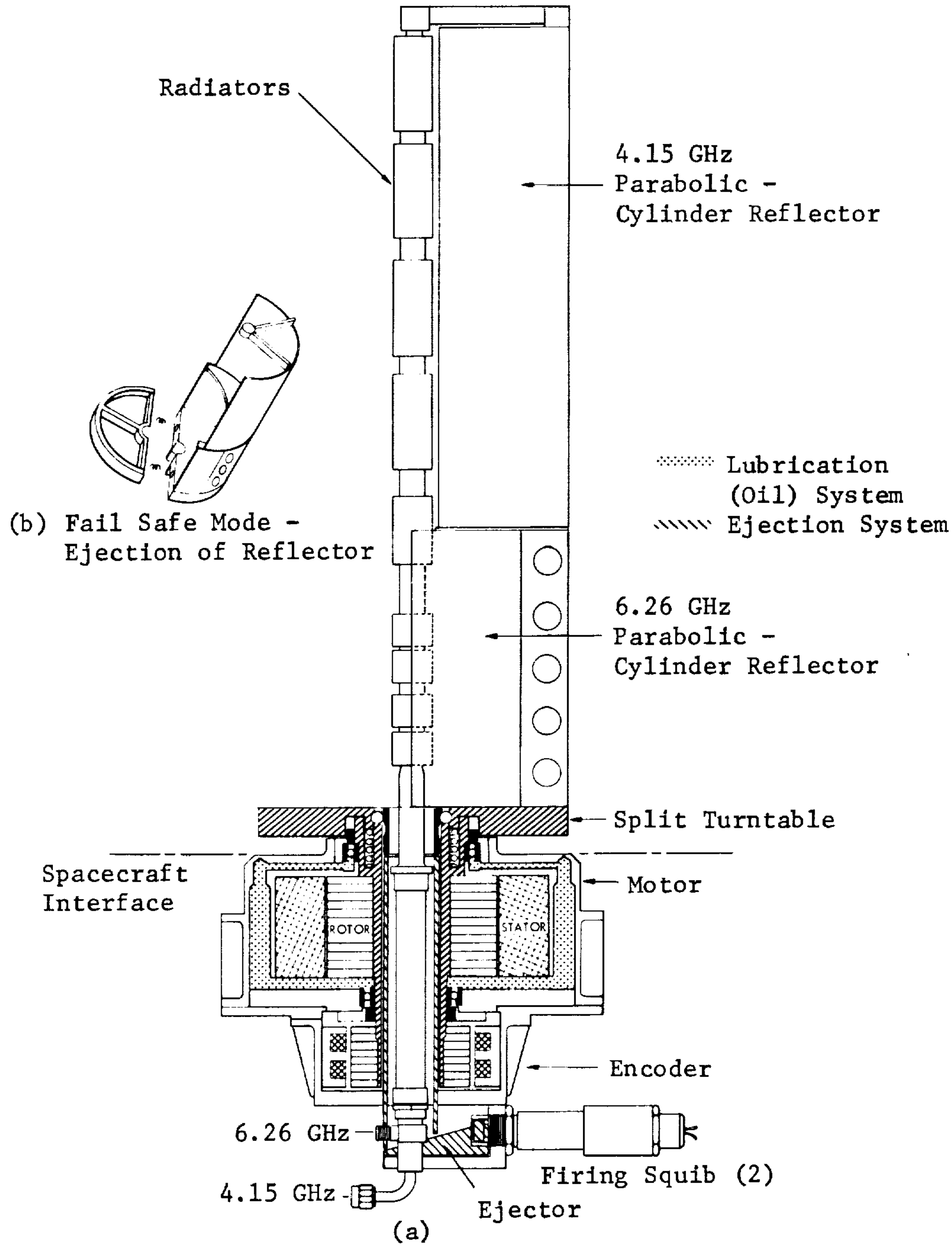


Figure 16 Antenna System for the ATS III Showing Oil Reservoir System

Courtesy of NASA Goddard Space Flight Center Greenbelt, Maryland

In situations where the bearing may constitute the hottest locality, lubricant evaporation will occur directly from the bearing. It could eventually result in a viscous residue. Hence, in these instances, where the condensation method could not apply, it is probably preferable to locate a separate reservoir next to each bearing so that its lubricant can be replenished by migration.

The extent, effectiveness and mechanism of migration is still conjecture. Reservoirs could be either stationary or mounted on the rotating element such that centrifugal force might play a part in the migration process. Much research needs to be accomplished in this area to enable design concepts to be engineered rather than guessed at.

Sintered Nylon or Nylasint has been an almost universal selection for oil reservoirs. It offers approximately 25% porosity when vacuum impregnated. Porus Polyimid is a new candidate with better dimensional stability. It is also easier to machine.

Figures 8 and 9 illustrate multiple uses of oil reservoirs used in conjunction with molecular seals. The arrangement of Fig. 8 has been operative for three years on OGO. In this case, the reservoirs are a contingency feature to maintain adequate lubrication in the event that the hermetic sealing failed. This explains the location of the "labyrinth shields." The lubricant is MIL-L-6087. The operating speed is 1000 rpm.

The antenna despin assembly of Fig. 9 embodies a labyrinthine molecular seal and four reservoirs to serve two bearings. Two of the reservoirs are stationary, two rotate. Lubrication is asserted to be by evaporation and condensation. This arrangement has operated satisfactorily for 3½ years so far on IDCSP/A. Operating speed is 100 rpm; the lubricant is wet vackote type 36194.

While the foregoing examples have been mostly low speed applications, the same basic arrangement has been used for smaller high speed motors.

2) Resevoir Type Retainers - Synthane (phenolic impregnated linen) and Nylasint (sintered porous nylon) have both been used for oil impregnated ball retainers. The former is available as a standard retainer material and is the normal recommendation for small high speed bearings. However, when considered from an oil retention capability, its capacity of approximately 3% is small compared to approximately 25% for the Nylasint and porous Polyimid.

While Synthane is considerably stronger than Nylasint, it exhibits considerable variation of properties and structure. Consequently, oil retention (which is largely a surface phenomena in the case of Synthane) varies widely. While in normal applications this is of little consequence, for space applications where the amount of oil is critical, it is very important.

The mechanism by which the oil impregnated retainer operates is alledged to rely on the force between ball and ball pocket, squeezing microscopic amounts of oil from the surface of the retainer. (Centrifugal force must also play a part.) The retainer is said to absorb equivalent amounts of oil from other contact points including the lands riding the race. Thus, a state of equilibrium is maintained.

According to MIT, the amount of oil needed by the ball/ball pocket interface varies with operating conditions and geometry. One important advantage of the sintered materials is that by varying pore size and pore distribution, the extent of lubrication can be adjusted; the amount of oil increasing as pore size is increased.

Hence, while sintered material is not as strong as phenolic, it offers the advantage of a much greater oil capacity plus the ability to regulate the extent of lubrication.

Under contract from the USAF, MIT is investigating porous polyimid for bearing retainers. In cooperation with the Dixon Corporation, they have developed methods by which pore size and pore distribution can be varied (as it can be in the case of nylasint). The polyimid material is more expensive than nylasint, but it is much easier to machine. Nylasint suffers from hygroscopic dimensional instability; it also has a memory which complicates machining. Deburring has to be accomplished in liquid nitrogen. Polyimid overcomes these difficulties and at the same time offers a much extended operating temperature range as compared to both nylasint and synthane.

Other research has been conducted with polimid of very fine porosity, but there have been accompanying reports of retainer instability problems. Such instability can result from excess friction due to inadequate lubrication at the ball/ball pocket interface or at the retainer/race-land interface. Very high hertz stresses are often involved causing severe retainer wear and high bearing friction, often increasing to serious proportions.

Presumably each different bearing application could demand a specific pore size; a requirement which would also have to be corrolated with oil viscosity and temperature range. Obviously, much research needs to be accomplished in this area before it becomes possible for an aerospace designer to select an oil impregnated ball retainer with assurance that it will meet the performance specification he has in mind. In the meantime, an empirical (cut and try) approach will be necessary commencing with an average pore size and pore distribution. As of this writing the Dixon Corporation appears to be the only manufacturer prepared to supply this material in a variety of pore sizes.

Unfortunately, the porous retainer represents such a small percentage of sales that the bearing manufacturers cannot be expected to underwrite the research and development necessary to corrolate all the variables with an optimum pore size.

Phenolic retainers are offered in single and two-piece configurations; the former being a race riding retainer; the latter a ball riding retainer with conforming ball pockets which provide better wipe-off of lubricant. With more ball surface in contact with the retainer, interface stresses are lower and, therefore, is supposed to involve less wear. This type of retainer is in use on certain Comsat despin bearings which are alleged to be experiencing instability problems. This type of separator involves a lower ball complement; and, thus, the aforementioned advantages have to be weighed against the disadvantages of fewer balls and an instability tendancy.

The improved stability characteristics of the single piece retainer is attributed to its race riding feature, but for this to be true it must be adequately lubricated since it is in fact a hydrodynamic bearing. Both outer and inner race riding retainers are available. The latter presumably would be preferrable with reservoir systems which feed the stationary outer race, in that the retainer does not impede oil migration into the bearing. However, this has not been proven. It may be that an outer race riding retainer might help to pump oil from the reservoir to the bearing. Compared to the twopiece retainer the single piece retainer has less pocket contact with the ball, stresses are higher and wear is said to be greater. However, the greater ball complement, which it can accommodate, will mitigate this.

It should be noted that the one piece retainer would be appropriate to the angular contact bearing (either single or duplex) since they utilize a greater complement of balls.

Another potential candidate for an oil retaining ball retainer is a material entitled polyoil (by Polypenco). SKF under contract to Comsat has performed limited development work on this. The material is highly porous and bonds directly to metallic ball cages. Thus, not only does it offer promise of an extended lubrication life, but the strength of the metallic cage is retained. Its current shortcoming is high torque--further development being necessary to solve this problem.

3) Grease Lubrication - A grease packed bearing (usually 30%) is the most compact type of oil reservoir. A channelling grease is to be preferred wherein the initial operation of the bearing will form a channel which will be fed with oil, bled from the residual grease. One problem with grease lubrication is that upon exhaustion of the oil content, a viscous residue remains causing excess friction. A possible method of engineering further life into grease lubrication might be to locate an oil reservoir immediately adjacent to the bearing in-lieu of one shield, with the grease in intimate contact and presumable absorbing oil as it bleeds oil into the raceway.

d) Breathing Provision - Some bearing failures from particulate contamination have been assessed as due to the air-flow to and from the motor housing during the course of preflight vacuum testing. This air-flow occurs through the molecular seal and through the bearing.

Bearing contamination results from debris within the motor being carried into the bearing during depressurization and from external contamination during pressurization. A similar situation will arise during Shuttle operations. When using molecular shaft seals, it is recommended that a breather be incorporated into the motor housing. It should consist of a 5 micron filter, large compared with the shaft seal in the laminar flow range, but of comparable size in the molecular flow regime.

4) Solid Lubrication Systems - There are two major functions for lubrication in space vehicle motors; the lubrication of bearings and the lubrication of gearing. Usually, the latter role is the more critical in that gear tooth engagement is essentially a rubbing mechanism with rolling only occurring at the pitch line, whereas ball bearings are regarded as pure rolling contact mechanisms. The former function can be accommodated by either dry film or transfer lubrication while the latter function is usually limited to the use of dry films.

A further function exclusive to solid lubrication is the brush used on high speed dc motors. (The slow speed torquer usually utilizes fluid lubrication--temperature permitting.)

a) Dry Film Lubrication - Solid lubrication is the only recourse when temperature extremes cannot be avoided. However, in the case of films they are not self-replenished; their life capability is limited, particularly at high rubbing speeds. However, they do excell under high load. Hence, low speed, high torque motor concepts should be selected whenever dry film lubes are essential. Dry films are used on both gearing and bearings, although bearings often utilize transfer lubrication in the form of sacrificial ball retainers.

Apart from the possibility of corrosion in humid atmospheres* there is no reason why dry film lubes should not be used for low operational life, long life applications; operational life (on gearing) being limited to the low hundreds of hours with the motor (fastest) pinion experiencing most wear. (Delrin or similar plastic pinions have been used successfully to extend pinion life--temperature permitting.)

In unhermetically sealed dc motors used in space vacuum, dry film lubrication should survive the life of the brushes, i.e., 50 to 200 hours operation. Applications may arise where this could prove an adequate and economic solution--as was discussed in Section A.1.f, the total, long-life operation time for motorized switches will amount to only a few minutes.

Ion sublimation deposition of MOS_2 gold and silver has been recommended as offering more tenacious and, therefore, more durable films. During Viking Surface Sampler development, the bonded MOS_2 film has proven to be superior. However the harmonic drive used in the biaxial antenna drive on DSCS 25 is an example where gold deposition has proven satisfactory. Hence, considerably more experience and research is required with this form of deposition before inconvertible recommendations can be made.

For comprehensive information on the wide variety of solid lubricants, reference should be made to Part A of the "Lubrication Handbook for use in the Space Industry," Control No. DCN-1-1-50-13616 (IF), available from the Office of Procurements and Contracts, D. E. Lyons, A&TS-PR-M, NASA, Marshall Space Flight Center, Alabama 35812.

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*Pertains to MOS_2 coatings and under investigation by NADRL.

b) Transfer Lubrication - The most commonplace bearing example of transfer lubrication is the Duroid retainer type of bearing. Duroid being glass reinforced teflon with approximately 5% MOS_2 impregnation. In some cases, Rulon, which is ceramic filled teflon, with MOS_2 addition has been utilized. The most recent innovation is the polyimid retainer.

In all such instances, the lubrication process is identical. The motion of the ball transfers microscopic layers of retainer material to races to form a dry lubricant film. The failure modes of this arrangement is excess transfer to the races and excess ball pocket wear. Both modes result from excessive ball/ball pocket interface forces which are related to ball velocity and load, and ball retainer velocity. The failure mechanisms are greatly aggravated by bearing misalignment.

Since ball load distribution is a function of bearing clearance and ball velocity is related to load distribution, it is advisable to use precision grade bearings and to provide axial preloading to unify ball load and velocity, thus minimizing ball/ball pocket interface forces.

The load ratings of these bearings are substantially below the normal L_{10} rating, the teflon retainer types particularly. Care must be taken to remain within these ratings. It is alledged that dry film lubrication of the races and balls improve the load rating capability of this type of bearing, but there appears to be no test evidence of this. It may be advantageous to longer life to apply the nonbonded type of dry film lube to the races of this type of bearing (e.g., Dicronite, Microseal and Vackote).

A ball retainer concept still in the experimental stage at WPAFB uses Boeing compact type 108 (designated AFSL 15 by WPAFB) for the entire retainer. This material which is primarily MOS_2 operates in the same fashion described for the Duroid retainer. A size 204 bearing having this form of retainer has successfully completed 32,000 hours of test operation at 1725 rpm with a 7 lb thrust load under 10^{-8} torr vacuum conditions. Presumably, this form of retainer would not be susceptable to the wear and wipe off problems described for the Duroid retainer. Considerable development lies ahead before this concept becomes a viable and available product.

Another 204 size bearing under test by the same laboratory uses a leaded Bronze retainer without any lubricant whatsoever. This has completed 40,000 hours of test. When last inspected at 1600 hours, very little wear was evidenced. Under the same test conditions, silverplated retainers failed at 300 hours.

Yet another development, where considerable testing is yielding very promising results is that by Elliot Brothers of Frimley, England, using lead coated races. Size 6-19-6 bearings have operated at 10^{-6} torr at 3000 rpm for up to 15,000 hours without failure.

While it would be premature to suggest the adoption of any of these three developmental bearing concepts, it is recommended that close cognisance be maintained; they might prove invaluable.

The use of solid lubricated bearings in induction motors should be decided with caution. The bearings must be relied upon to convey the majority of the induced armature heat to the housing; the poor heat conduction of dry filmed ball contacts may be very inadequate causing excessive heating, differential expansions, loss of bearing clearance, severe wear and failure from excessive friction.

Transfer lubrication of gears has been accomplished using sacrificial idlers of Duroid or similar material having an MOS_2 content. The meshing action of the teeth perform the required wiping action.

A similar technique can be utilized with the Boeing compact materials (licensed to Pure Carbon) using sandwich construction.

Solid lubricants operate within the realm of boundary lubrication. They should be resorted to for long life applications only when fluid lubrication becomes impractical. They function as beneficial contaminants between mating surfaces and do not prevent metallic contact. They merely alleviate the wear problem, whereas the intent of fluid lubricants is to prevent metallic contact. When a fluid film deteriorates to the point where boundary lubrication is evident, it is looked upon as the onset of a failure mode. Dry film lubricants, on the other hand, actually operate within this failure mode. Hence, it can be appreciated that the life of dry film lubricants is short compared with fluid film lubrication.

For long term Shuttle type programs, a servicing and replacement program must be instituted for motors when the use of solid lubrication is unavoidable and total life requirements exceed the capability of the solid lubrication system.

b. Results of Survey - Four space equipment manufacturers were invited to submit their opinions as to why certain successful motor applications proved so successful. Subsequently the suppliers of such motors were requested to identify reasons for this success. This information is presented in Table 3.

c. New State of the Art Research Recommendations - The recommendations follow:

1) Correlate and co-ordinate bearing and lubrication research activities throughout the USA by a component authority. Total effort and expense (usually funded by government agencies) would decrease with a concerted plan. It would not only avoid unnecessary duplication of effort, but yield systematic data increments of maximum value to the development of bearing/lubrication theory and technology.

2) Research the mechanisms of oil reservoir operation: can judicious positioning of reservoirs adjacent to bearings promote oil migration in addition to the alledged evaporation/condensation process? It is submitted that the reservoir oil content could be dyed to track oil excursion. For larger bearings, rotating reservoirs could be used where centrifugal forces could promote migration; these affects should be investigated and documented. Do reservoirs, in fact, provide any of the advantages attributed to them? Some authorities believe that lubricant coating of all interior surfaces of a bearing or motor housing is the fundamental requirement. Is this true?

3) Research the mechanism of lubricant loss through molecular shaft seals and associated filters: does liquid creepage occur under certain conditions?

4) Continue research into the evaporation rates and volatility fractions of selected "space" lubricants, for more accurate prediction of vapor loss from molecular seals, improved sizing of oil reservoirs and better insight into how lubricant films change with time.

Table 3 User/Manufacturer Survey

PART DESCRIPTION AND USING PROGRAM	USER'S OPINION OF WHY PART WAS SUCCESSFUL	MANUFACTURER'S OPINION OF WHY PART WAS SUCCESSFUL	RECOMMENDED GUIDELINES IMPROVEMENTS, PROCUREMENT SPEC REQUIREMENTS, ETC.	RELATIVE IMPACT YIELD & COST OF INCORPORATING RECOMMENDATIONS
A. Reaction Wheel Induction Motor & Bearings for Model 35 Defense Satellite	Special care given to bearing & lubrication system using V79 Esso oil impregnated ball retainer plus Andoc C grease.	1) The use of twin duplex bearings, one rigidly mounted in housing, the other mounted in a cartridge which slides in the housing to accomodate differential thermal expansion or contraction. 2) The use of basically ABEC 7 bearings with certain dimensions held to better than ABEC 9 tolerances. 3) Vacated wheel housing, thus requiring only 6 watts of power. 4) Compatible insulation throughout motor, i.e., the magnet wire, the impregnant and slot insulation were selected to avoid chemical reaction. Also stringent process controls.	For longer life an improved reservoir material would be used. This is a proprietary combination of material and processing.	insignificant

Table 3 (cont)

PART DESCRIPTION AND USING PROGRAM	USER'S OPINION OF WHY PART WAS SUCCESSFUL	MANUFACTURER'S OPINION OF WHY PART WAS SUCCESSFUL	RECOMMENDED GUIDELINES IMPROVEMENTS, PROCUREMENT SPEC REQUIREMENTS, ETC.	RELATIVE IMPACT ON YIELD & COST OF INCORPORATING RECOMMENDATIONS
B. Despin Motor & Bearing Assembly for Tacomsat using Inland Motors Torquer Model T5151.	1) Low Speed; 33 RPM 2) Special Cartridge type brushes 3) Lubrication System using molecular shaft seal, Nylasint reservoir & high porosity (4%) synthane ball separators in ball bearings. 4) ABEC 7 bearings with grade 10 balls 5) Apiezon oil with lead napthalate (Bardahl) i.e., Vackote or Hackote.	Manufacturer cited the following special processing: 1) High Temp HML magnet wire. 2) Hi Film, Kapton Polyimid Slot Insulation. 3) Special, high accuracy Commutator Bar Spacing 4) Special deburring and inspection of commutator under magnification. 5) The use of cartridge brushes with negator springs.	As far as the motor is concerned, extended life is enhanced by curtailing current levels well within the temperature rating of the insulation and brush capability, the latter minimizing the rate of degradation. Also the cartridge type brushes can provide many times the life of their standard cantilevered brushes. For dependable bearing operation, high quality bearings ABEC 7 with grade 10 balls are necessary with properly impregnated, high porosity (4%) Synthane ball retainers.	No Cost Impact 50% Increase No Cost impact since these are the logical choice in any case.

Table 3 (cont)

PART DESCRIPTION AND USING PROGRAM	USER'S OPINION OF WHY PART WAS SUCCESSFUL	MANUFACTURER'S OPINION OF WHY PART WAS SUCCESSFUL	RECOMMENDED GUIDELINES IMPROVEMENTS, PROCUREMENT SPEC REQUIREMENTS, ETC.	RELATIVE IMPACT ON YIELD & COST OF INCORPORATING RECOMMENDATIONS
C. PM Stepper Motor for Hermetically Sealed Antenna Drive (Wabble Gear) as used for OGO & Nimbus IV. See Fig. 15	1) Unit was Hermetically sealed and charged with an atmosphere of GN_2 2) Motor operates at a derated load & derated speed (3000 RPM) 3) Special attention to heat sinking. 4) Special attention given to lubrication: Gears are grease plated, motor bearings have phenolic, HT102 oil-impregnated separators. A nylasint oil reservoir is also incorporated.	There are no isolated reasons. It is a matter of overall expertise acquired through long experience and exact procedures. For instance, another division of the same company working to the same drawings can fail to produce a satisfactory product.	They (the motor manufacturer) would prefer to "design from scratch". Instead for economy reasons, new requirements are usually met by modification of existing designs - As a result, motors are often over-designed.	100% to 200%

Table 3 (concl)

PART DESCRIPTION AND USING PROGRAM	USER'S OPINION OF WHY PART WAS SUCCESSFUL	MANUFACTURER'S OPINION OF WHY PART WAS SUCCESSFUL	RECOMMENDED GUIDELINES IMPROVEMENTS, PROCUREMENT SPEC REQUIREMENTS, ETC.	RELATIVE IMPACT ON YIELD & COST OF INCORPORATING RECOMMENDATIONS
D. Induction Motor (complete with bearings) for Horizon Scanner as used on Agena (and other vehicles). See Fig. 7.	"Sheer Luck - for instance if I ask 8 experts advice, all 8 opinions are different."			
	NOTE: In this instance user and manufacturer were the same corporation.			

5) Load & Life capability of Porous Polyimid Retainer Bearings should be explored beyond the realm of gryoscope bearings as currently being investigated by MIT. Concurrently, the mechanism of lubricant flow from and back into the retainer should be investigated, as well as retainer stability and wear phenomena.

6) Evaluate Boron deposition coatings to 52100 alloy steel bearings to provide non-corrosive properties superior to 440C. The objective is to obtain the superior life capability and the advantages of the 52100 steel.

7) Investigate the false brinnelling hazard of the severe vibration environments of the Shuttle program where very high G levels are maintained for 5 hours/axis. Typical state of the art motor and bearing technology should be evaluated under realistic vibration conditions to assess the actual areas where state-of-the-art improvements are needed.

8) Extend the exercises of 7) above to all types and facets of motor design and construction.

9) In conjunction with items 7) and 8) above, review the necessity for special bearing preloading measures indicated in Figure 13.

10) Corrolate and publish all visco/pressure data. Assess where additional research is needed in this field to facilitate the design of bearing systems for space. Fund and implement this additional research. (It will be noted that the voluminous "Lubrication Handbook for use in the Space Industry" omits this important information.)

11) Implement research into the necessary extent of lubricant contamination control for satisfactory utilization in space mechanisms where thin films with minimal replenishment predominate.

12) Implement a system for recording motor operational history in space in order that factual life data on bearings and lubrication systems is conveniently available from which the relative merits of different systems can be evaluated and from which the necessary direction of research can be established.

13) Explore new state of the art technology better capable of resisting shuttle environment and life requirements. This is to include not only the evaluation of the more sensitive areas such as bearings,* but also dormant constructional features such as windings, conductor terminations, connectors, lamination stacks and rotor configurations.

14) Perform experimental investigation of radially preloaded or zero clearance ball bearings. They could accommodate the differential thermal expansion and contraction of the motor rotor with respect to the housing, at the rolling element, thus eliminating sliding bearings. The effect of their spring rate on motor resonant frequency also needs investigating as a means of negating vibration affects (false brinnelling).

15) Conduct a comprehensive test evaluation program on reinforced retainers to provide proper load/life recommendations. Bearings with ceramic or glass reinforced Teflon retainers (with approximately 5% of MOS2 added), have been widely used with considerable success in many space applications. However, their load ratings are considerably reduced from normal and their life capability is not documented. According to certain sources, the application of dry lubricant film to balls and races substantially improves the load/life capability of such bearings. However, there appears to be little or no test data to corroborate this. Designers have to work with very inadequate bearing performance data when contemplating the use of this type of bearing.

16) Mechanical Signature Analysis is a technology which can diagnose the mechanical health of mechanisms without their disassembly. It would therefore constitute an important means of effecting economics in the cost of maintenance and inspection on Shuttle type vehicles. Its use necessitates that a library of mechanical signatures be assembled for each of the components to be inspected. These would be acquired during qualification and development testing and represents an additional cost which must be assessed in relation to the cost economies resulting from the avoidance of unnecessary maintenance. The extent of such economies and the ramification of instituting such a system should be investigated.

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*For instance the spiral groove plain (sleeve) bearing, which is being evaluated for use on gyros and reaction wheels may be beneficial in ordinary motor applications. See References 8 and 9.

The following "Recommended Areas for Study" are extracted from NASA Report No. TmX-67872, entitled *Lubrication Friction and Wear in Aircraft*. This study confirms some of the above recommendations and presents several other basic areas of research.

RECOMMENDED AREAS FOR STUDY (from NASA Report TmX 67872)

1. <u>The Solid and Its Surface</u>

   a. Definition of surfaces from atomistic to macroscopic scale of observations (includes computer program analysis of surfaces in contact).

2. <u>Dry Sliding Contact</u>

   a. Effect of small alloy additions on adhesion
   b. Development of carbon-graphites for brakes

3. <u>Solid Lubricants</u>

   a. Development of solid lubricants for operation to 871°C (160°F); also solid lubricants to 1371°C (2500°F).
   b. Investigation of wear mechanism on atomistic scale.

4. <u>Liquid Lubricants</u>

   a. Rheology of lubricants at high pressures and shear rates.
   b. Synthesis and formulation for high temperature and improvement of fluid film rheology.
   c. Additive and surface chemistry to control corrosion and dirt and improved boundary and E.P. films.
   d. Investigate friction polymers.

5. <u>Lubricant Systems</u>

   a. Methods to improve cooling (includes techniques for improved application of lubricant, mist lubricating, and separate cooling and lubricating flows).
   b. Improved filtration
   c. Controlled atmosphere

6. <u>Elastohydrodynamic Lubrication</u>

   a. Investigation of entrapment and normal approach effects
   b. Differences among various lubricants

7. Gears

    a. Improved scoring model
    b. Review of the effect of gear errors on lubrication
    c. Investigation of run-in effects on surface and on profile geometry.

8. Bearings

    a. Information on differences in fatigue life between case carburized and through-hardened materials.
    b. Improved method for predicting bearing life taking into account material, lubricant and film thickness effects.
    c. Reduced stress at high speeds.

9. Seals

    a. Improved abradable material and a model for predicting abradability property.
    b. Seal carbons with improved erosion and oxidation resistance.
    c. Materials for reducing secondary seal fretting.

Also, it is recommended that support be lent to the IRG (OECD) group in work on:

    a. terms and definitions (and translations)
    b. thermodynamics of failure.

d. Hardware Life - The actual life data of Table 4 was obtained from the MMC *Survey of Motors Used in Space*, report No. T-71-48890-004. Unfortunately, the establishing of factual operational life data of motors in use on satellites is very time consuming. Truly reliability sources are difficult to locate. Usually if there are no problems, there are no records.

In view of the significance of past accomplishments in determining the appropriate areas of new research, it is submitted that a more organized approach to record keeping be implemented, not only for motors but for many other important mechanisms.

Table 4 Examples of Motor Life Capability in Space

TYPE		RPM	LIFE
Hysteresis Synchronous (No Gearhead)	OAO Tape Recorder	8000	One Failed at 3 years
Induction Motor (No Gearhead)	OGO Reaction Wheel	1500	Still Running in Space at 3.5 years Still Running in Lab at 8 years
Induction Motor (Gearhead)	Nimbus III Solar Array Positioning	8250	Still Running in Space at 3.5 years
Brush Type Torquer (No Gearhead)	OSO III Despin	30	Still Running in Space at 4.6 years
Brush Type Torquer (No Gearhead)	ITOS I Momentum Wheel	150	Failed in Space at 0.6 years Commutator Clogging Redesigned to Brushless DC Motors
Brushless DC Torquer	IDCSP/A Despin	100	Still Running in Space at 3.5 years
PM Stepper 90 Step (No Gearhead) 100 PPS	ATS Spin Scan Camera	1500	Still Operating in Space at 5 years

5) Bearing Selection - Section A offers a derating system which should enable bearings to be selected to give high long-life reliability under favorable lubrication and installation conditions. It was also shown that precision surface finishes enhance lubrication condition and life expectancy. In Section C2a3c, the merits of certain types of bearing retainers were discussed. In Section C2a2, preferences were established for preloading systems to acquire specific bearing stiffnesses.

In regard to duplex bearings, high precision bearings will be necessary to hold their preload value and/or stiffnesses within reasonable tolerances. Contact angle and raceway radius will have to be tailored to suit. The diametral interference of the bearing in its housing will influence preload. This must be checked and shimming may prove essential. ABEC 7 precision grade bearings with 1½ microinch raceway surface finishes and balls of 0.5 to 1.0 microinch finish should be used.

The most suitable ball retainer depends on many factors. For instance for flooded pump motors, standard stainless steel may be preferable. Porous polyimid is recommended for bearings which must rely on oil reservoirs and oil impregnated retainers, selecting the porosity to suit oil viscosity and temperature range.

Retainerless (full ball compliment) bearings should not be used--except possibly when maximum capacity is needed in restricted space at low speed. In marginal lubrication conditions, excessive ball on ball wear has been experienced with retainerless bearings.

In the case of solid lubrication, the Duroid retainer (MOS_2) has provided exceptional service under light load; but there is no body of test data from which a definite load or life commitment can be determined. Also, as mentioned in Section 2a4b, there are numerous solid film lubrication concepts under test apparently capable of very substantial life. Presumably, one of these could be an optimum choice. However, information on these are not generally available to the designer. Also, the extent of test data pertains to only one or two sizes of bearing. It is probably risky to extrapolate this to other bearings.

For maximum benefit to be gained from all these independent test programs, an authority should be delegated to correlate the data and publish application recommendations. Test programs should be intensified and broadened in accordance with an overall systematic plan and application data enlarged and updated as testing yields results. Greater opportunity should be extended to commercial bearing houses to participate in this activity in addition to government and military establishments.

In regard to bearing materials, there should be no need to resort to the special tool steels. These are advantageous in high temperature applications in that they offer excellent hot-hardness characteristics. But they introduce difficulties in regard to surface finishing--in any case this study is not concerned with very high temperatures.

Since 52100-consumable-electrode-vacuum-remelted-steel is still regarded by some authorities as offering improved life and fewer inclusions as compared to 440C produced by the same vacuum remelting method. 52100 should be utilized for long life applications when there is no risk of corrosion. Otherwise, 440C should be used.

Means of overcoming the corrosiveness of 52100 would be of special advantage to the space industry as well as to industry in general. A possible solution of this problem is Boron deposition coatings.* Platings in the past have proven unsatisfactory, however; preliminary investigation of boron deposition (which is not a plating process) indicated that it can withstand normal race wear without losing its corrosion inhibiting properties. Therefore, in the interest of long-life space requirements, it is submitted that the investigation of this process for bearings may be a very rewarding avenue of research.

Another process which proved advantageous in boundry lubrication was soaking of balls and races in tricresyl phosphate (TCP) (Ref 10.)

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*This suggestion and permission to publish by courtesy of SKF Industries.

Bearing surface chemistry has been shown to be an important factor governing bearing performance under fluid lubrication. Contamination can adversely affect surface wettability. Nonwettability can result from contaminated solvents, cleaning water, containers and white room atmospheres also, certain solvents and detergents can create surface contamination. Meticulous housekeeping controls and processing procedures must be followed. Reference 11 reviews this problem area and defines a wettability test procedure for bearing races.

6) Bearing Installation - Obviously, the ultimate in quality engineering is expected of spacecraft hardware. Thus it is not intended to review the normal precautions which must be carefully checked in respect to bearing installation--such as proper radii at shaft shoulders and in housing bores. Nor should it be necessary to emphasize the need for worse case analysis in order to determine the tolerances necessary to limit misalignments and eccentricities to acceptable levels.

What should be investigated are the special installational measures needed to accommodate the frequent repetition of environment extremes which will often accompany long life missions. The effect of space vacuum on bearings is largely lubricant oriented and this is covered in Section C2a3. Vibrational problems in regard to bearings are covered in Section C2a2, which covers certain installational recommendations.

It is temperature variation which poses the main problems for bearing installation. There are two main aspects of this problem (1) linear differential expansion and contraction of the rotor assembly relative to housing, and (2) radial differential expansions and contractions. Furthermore this problem is not only due to the effect of ambient temperatures changes on different materials, but also, due to differential temperatures caused by heating of electrical windings during periods of motor operation.

a) Linear, Differential Expansion or Contraction - The bearing installation problem which results from this temperature dependent phenomenon is that of avoiding excess thrust on the bearings. The most elementary solution is to allow one and sometimes both bearings to "float" longitudinally in the housing. However, in order to float in this manner, the bearing must also have radial clearance with respect to the housing. While this end float and radial clearances overcome the temperature dependent problem, it becomes very vulnerable to vibration. The vibration can cause hammering

of bearings in the housing with wear and galling of the housing at the bearing O.D. resulting in possible jamming of the bearing, brinnelling of the bearing races and wearing of the ball retainer. Hence, this design approach should only be used for very mild vibrational environments.

These shortcomings can be largely overcome by restraining axial movement of the rotor with respect to the housing by using angular contact bearings, one at each end of the rotor with a spring at one end which preloads both bearings. However, one bearing must still be allowed to float under the action of the spring (usually a Belville washer) as temperature fluctuates. This arrangement has successfully qualified for many space programs. However, for really long-term vibration requirements, the spring resonance, which reacts with the full rotor mass, could be a liability causing fretting corrosion between the floating bearing and housing and false brinnelling of the raceways. See Figure 6a.

These problems can be alleviated by using a preloaded duplex bearing at one end as shown in Figure 6b, which physically prevents axial movement of the rotor. However, the bearing at the other end of the rotor must still float relative to the housing; but it is now divorced from the mass of the rotor (during vibration and shock). While the inner race of this floating bearing is an interference fit on the rotor shaft, the outer race must be free in its housing, the extent of this freedom being temperature sensitive. This freedom contitutes a vibration hazard. It can be minimized by using an insert in the housing (could be termed a compatibility bushing). The insert is machined from metal with a temperature coefficient of expansion equivalent to that of the bearing. Inserts should be an interference fit in the housing and of sufficient cross-section that the differential thermal strain will occur predominantly in the housing. Bearing clearances can still constitute a false brinnelling hazard. It is, therefore, advantageous to use a preloaded angular contact bearing with the preloading spring also accommodating the linear differential expansion and contraction.

The use of the duplex bearings at both ends of the rotor (one fixed, one floating), combined with compatibility bushings, is the recommended approach when precision air-gaps and molecular seal clearances are demanded. This arrangement avoids preloading springs altogether as illustrated in Figure 6c.

However, as previously discussed, the resonant frequency of the floating bearing and its preloading spring (if used) can still represent severe vibration problems; in which case, the arrangement shown in Figure 13 should be considered. The primary advantage of this arrangement is that the floating bearing is eliminated. Instead, the rollers or balls slide relative to one of the races. In mild vibrationary conditons where stiffness of the bearing may not be important, a zero clearance version of this configuration could be adopted.

The problem of thermal expansions and contractions could be largely eliminated if the housing could be of a steel with similar characteristics to those of the rotor shaft, usually a steel alloy. However, this is also a problem in that usually the housing needs to be non-magnetic. This limits the choice to the 300 series (austinetic) stainless steels, but these have poor heat transfer characteristics and unsuitable temperature coefficient of expansion. Thus, steel does not constitute a satisfactory selection for motor housings.

A review of materials shows that the only non-magnetic materials with compatible temperature coefficients of expansion are Titanium and Beryllium. Of these, the Titanium has poor heat conduction characteristics, Beryllium has good heat conduction characteristics. Titanium offers considerable cost economies as compared to Beryllium.

Hence, motors for use over wide extremes of temperature should have Beryllium housings. Motors involving short or intermittent duty cycles and little heat dissipation, can utilize Titanium. (Titanium avoids the machining complexities of Beryllium and should be less expensive.)

Coefficients of Expansion of Candidate Materials for Motor Housings are:

Beryllium Alloy	6.4×10^{-6} inches/°F
Titanium Alloy	5.2×10^{-6}
CRCS 300 Series	10.4×10^{-6}
CRCS 440 C	5.6×10^{-6}

Alloy & Carbon Steels	6.5×10^{-6}
Aluminum Alloy	12.0×10^{-6}
Magnesium Alloy	13.0×10^{-6}

Bearing bores in housing must be circular to extremely tight tolerances, otherwise the bearing will be distorted, resulting in eratic behavior. Housing should be designed as symetrically as possible to avoid distortion with change in temperature, for the same reason. Suitable stress relieving procedures should be adopted before and during the course of machining. This is particularly important in the case of thin section bearings which rely on the housing to correct their inherent ovality.

When operating temperatures depart substantially from the temperature at which the motor operates and particularly if a differential temperature exists between inner and outer races, the "fit" of the bearing both on the shaft and in the housing becomes extremely critical, especially when thin section bearings are used (which is often the case in aerospace components in order to achieve low weight). The extent of this fit determines the interface stresses and strains. It also influences the differential thermal strains of both housing and bearings, which will contribute to the installed geometry of the bearing, influencing clearances, frictional torque, heating, spring stiffness and film thickness.

Hence, catalogue (AFBMA) recommended fits can only be used as a guide in many circumstances, actual fits must be carefully computed. It may be necessary to custom code the bearing and selectively assemble them in appropriate housing bores. This problem area can be alleviated by the use of either housing materials or compatibility bushings with temperature coefficients of expansion akin to that of the bearing.

d. *Application Guidelines* - The guidelines are discussed in the two following subsections.

1) *Simplicity, Low Speed, Sturdiness of Construction and Heat Flow Paths* - In general, the motor should drive as directly as possible; i.e., with minimum gearing interposed between motor and load. This will often mean that a larger diameter, slower speed, higher torque motor is utilized. This may, in turn, incur

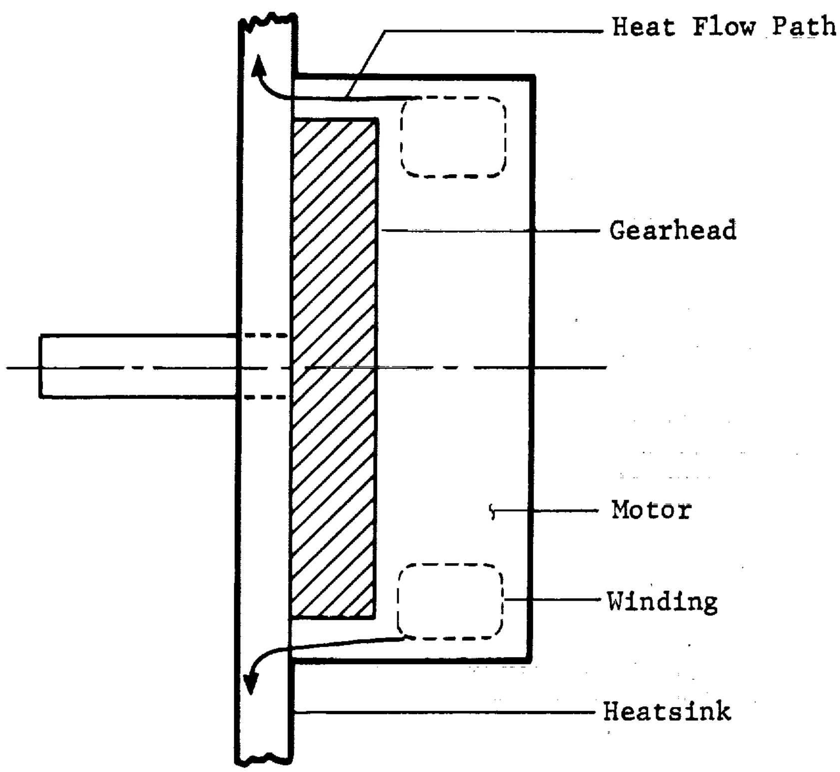
a weight penalty; but it will mean that a more sturdy (lower L/D ratio) motor is used. This philosophy can be extended using multi-pole configurations until an essentially pancake type of motor is involved.

The inherent advantage of this approach is that fewer gear stages will be needed, or if the concept is taken to its ultimate conclusions, all gear elements will be eliminated and the motor will drive directly. Figure 8 is an actual example where this type of motor superceded the conventional type during the course of product development to achieve longevity. Motor efficiency in this circumstance may not be optimum; but this is somewhat offset by the avoidance of gearing in efficiencies and mechanical complexity. Furthermore, a higher power drain may be of lesser evil than the hazard of high speed gearing and many bearings, particularly if dry film lubrication is demanded by the environment.

Figures 1 and 17 demonstrate the relative advantages of the slow speed pancake configuration. Not only are fewer gear passes required, but what gears are needed can be designed sturdily with adequate tooth sizing, conservative stress levels, low pitch line velocities and absence of undercutting. Usually, the fewer bearing there are, the better the reliability. It must not be construed that high speed motors in themselves are a hazard, in fact it is well known that high speed gyro motors operate for many thousands of hours with minimal lubrication but, if gear reduction is involved, the slowest motor is advantageous--except possibly from a weight point of view.

Another advantage of the pancake configuration is that the heat of the winding is directly conveyed to the heatsink through the casing of the motor. Whereas, in the case of the conventional (cylindrical) type motor, the heat is conducted through the gear-head, thus aggravating lubrication and bearing problems in that area. In comparison, the cylindrical motor has a longer, less substantial heat flow path which, unless augmented, may well cause a greater temperature differential; i.e., a hotter motor. See Figure 18.

A still further advantage of the pancake configuration is the inherent rigidity of mounting, an attribute of particular importance to combat severe vibrationary environments such as those of the Shuttle Program. (It is recognized that many applications will arise where the conventional motor, because of better availability, or lower cost, or even on technical grouds, will be the logical choice and will satisfy long life specification, particularly in the low duty cycle and/or benigh environment situations.)

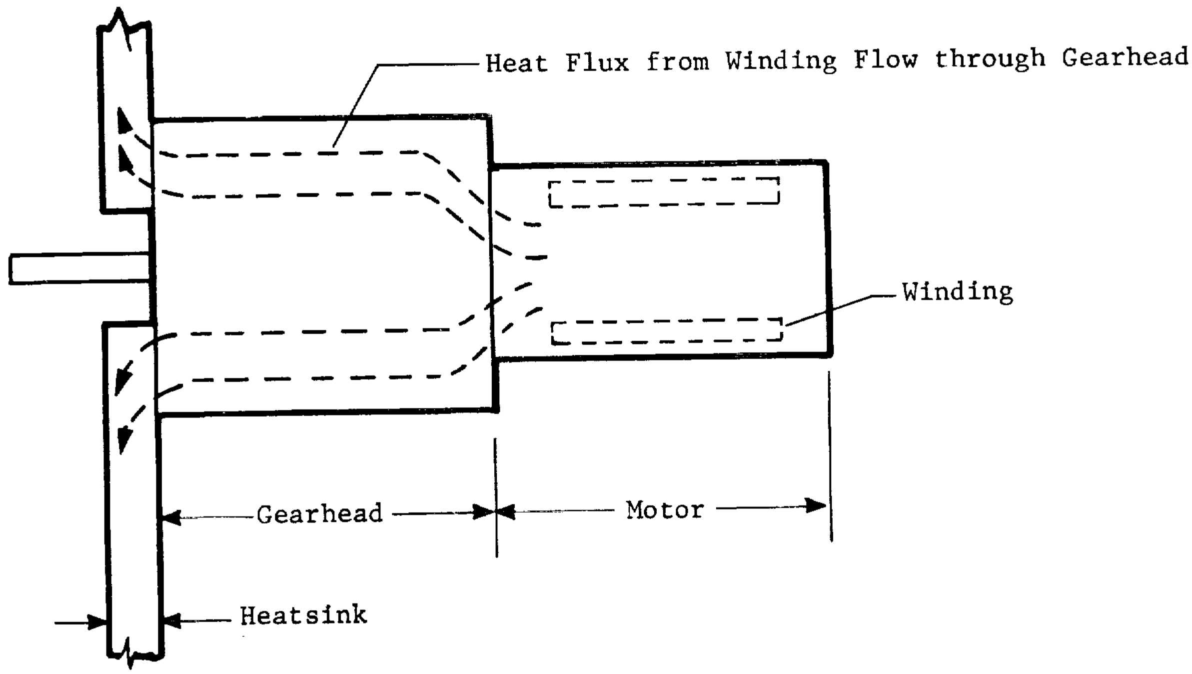


Pancake Type Motor

Features

- Low Speed-High Torque
- Low Speed Gears - Lower Sliding Velocity
- Avoidance of High Speed Bearings
- Heat Flux By-Passes Gearhead
- High Structural Rigidity
- High Stiffness of Rotating Ass'y (High Natural Frequency)
- Low Gear Ratio ie Fewer Gears and Bearings

Figure 17 Pancake Type Motor



Conventional (Cylindrical) Type Motor

<u>Features</u>

- High Speed - Low Torque (Very Sensitive to Frictional Changes)
- High Speed Gears (High Wear)
- High Speed Bearings (Life Limited when Dry Film or Marginally Lubed)
- Heat Flux Flows through Gearhead (Higher Temp')
- Low Structural Rigidity (More prone to vibration and shock problems)
- High Gear Ratio (Multiple Gears & Bearings)

Figure 18 Conventional Type Motor

It is again stressed that to acquire maximum benefit from all these attributes, the motor must be integrated into the driven mechanism and not merely added to it as an afterthought.

Figures 7 thru 10 and 13 are good examples of how this can be accomplished. In some instances, the rotor can be advantageously installed on an existing rotating assembly, thus avoiding the need for separate motor bearings.

This same philosophy extends to high speed motor applications where the motor drives directly, i.e., without gearing. Examples are mirror and chopper drives for optical instruments. These amount to ideal applications in that the high speed facilitates full EHD operation of the bearings. However, there is no justification for the use of excess gearing on a slow speed application, merely to facilitate the use of a high speed motor. As previously mentioned, the motor should drive as directly as possible so as to minimize gearing and bearings; this applies regardless of whether the output shaft bearings operate in the EHD or boundary regime.

2) *Motor Output Shaft-Coupling Methods* - Often it is desirable to connect the motor to the driven mechanism by means of gearing, probably with a step down ratio with a pinion mounted on the output shaft. In such arrangements, it must be realized that the pinion tooth load is reacted at the centerline of the motor shaft causing additional loads on the motor bearings. While this represents very elementary mechanics, it is surprising how often this additional bearing load is ignored. Furthermore, the smaller the pionion, the higher the load; however, if the gearhead is of the offset type, it could add or subtract to bearing loads according to the relative angular position of internal and external tooth meshes. Hence, the motor manufacturer must be informed of the applied loads so that he may select bearings of appropriate capacity--see Figure 19a.

Also, when designing the motor pinion, the stiffness of the motor shaft must not be overlooked when calculating the effect of misalignment on tooth stresses. For a given motor torque, the larger the pinion, the lower the tooth load; and thus, the additional bearing loads will be diminished. The use of a relatively large pinion is therefore recommended when this method of torque transmission is adopted. Also, the pinion should be cantilevered as little as possible, i.e., located as close to the motor as practical. Figure 4 illustrates a judicious selection of pinion for the motor.

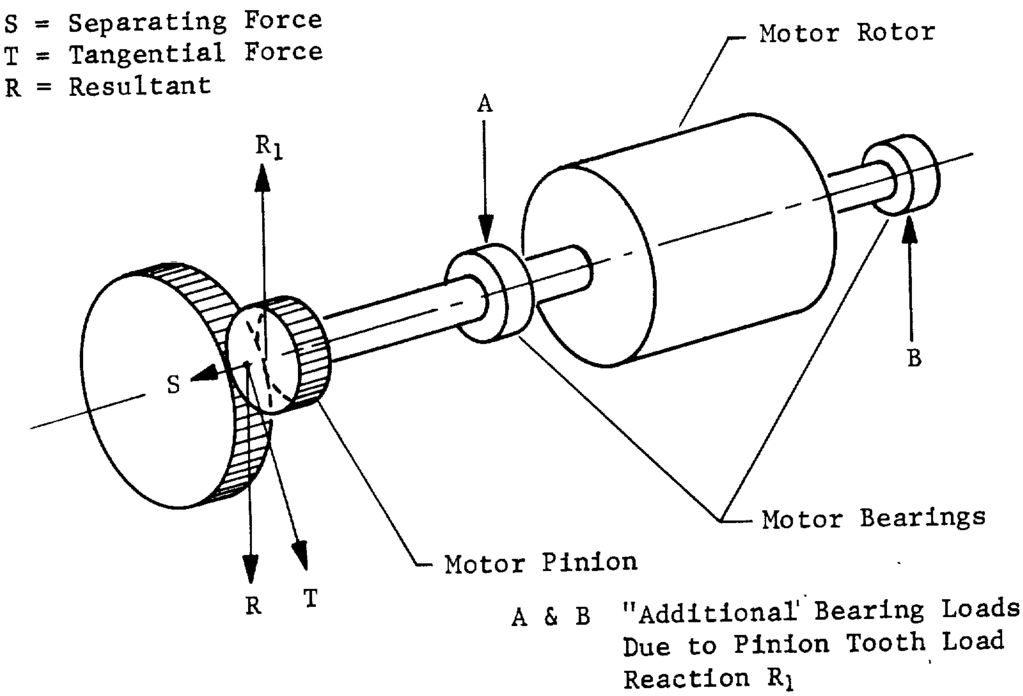


Figure 19a Unbalance Motor Pinion Tooth Loads

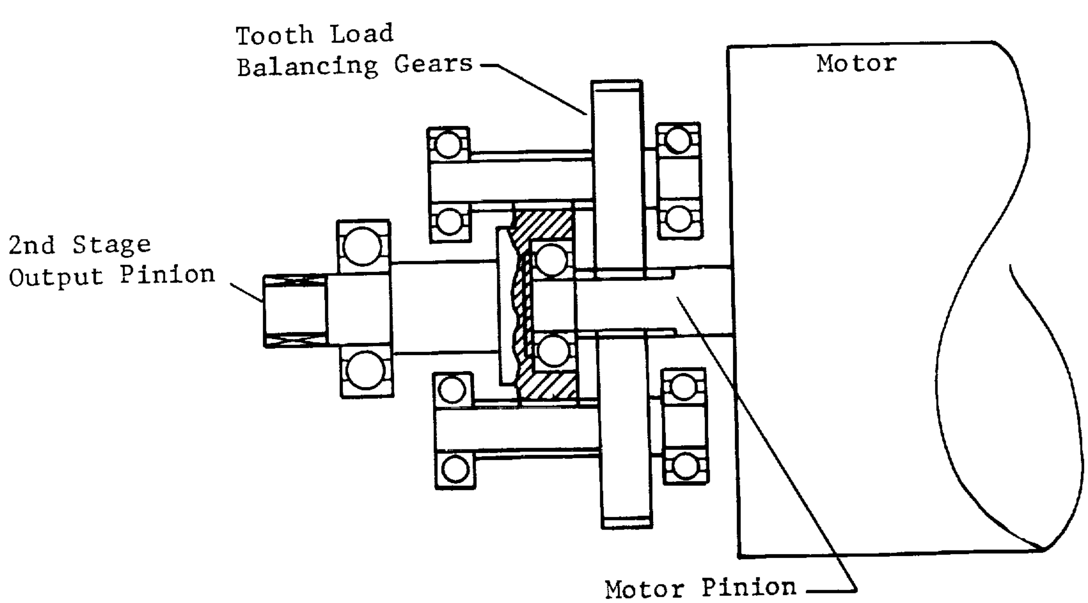


Figure 19B Balanced Motor Pinion Tooth Loads

Splining the output shaft into the driven shaft should theoretically avoid the additional bearing loads previously mentioned; but in practice, due to unavoidable errors in alignment, some loads will be imposed. The ideal method of transmitting motor torque in this axial fashion is to use a quill; see Figure 20. Alternatively, flexible couplings will also avoid the imposition of additional loads on the motor bearings--if properly applied. These methods, more commonly pertain to larger size geared motors.

A further method of avoiding these additional bearing loads is to adopt the arrangement of Figure 19b which balances out the tooth loads. This is more commonly used as the first stage of a gearhead to alleviate pinion tooth load, but it also reduces the motor bearing loads. It doubles the number of gears and bearings in the stage, but tooth loads are halved. However motor pinion experiences twice the number of tooth engagements. Thus the arrangement is likely to increase pinion wear, particularly in the case of dry film lubricants at high speed.

Note: All the preceding application guidelines pertain to motors. Application guidelines for bearings are covered in various sections of C2 where Vibration, Lubrication and Installation problems are discussed as a facet of motor technology.

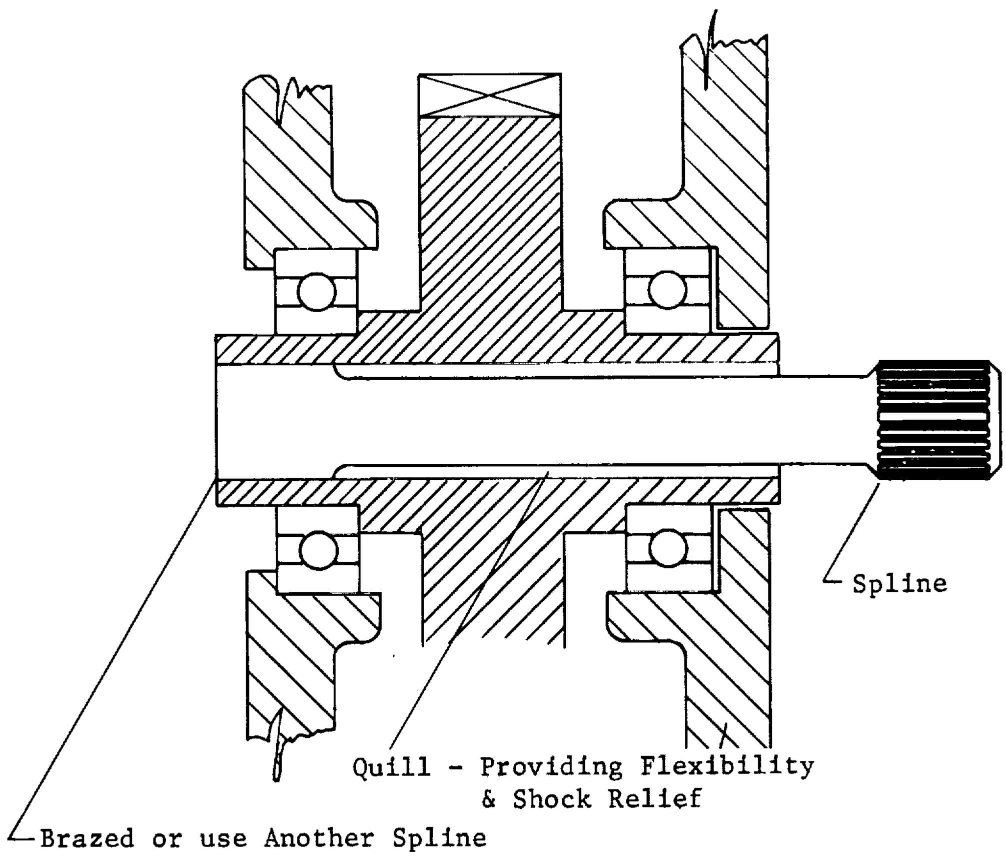


Figure 20 Final Output Stage Showing Splined Quill

D. TEST METHODOLOGY AND REQUIREMENTS

1. Qualification

Qualification of motors must encompass the full variation of bearing geometry as determined by bearing manufacturing tolerances, installation tolerances plus any other significant variables such as preload, spring stiffness, contact angle, misalignment, eccentricity and temperature extremes.

Multiple qualification samples should be carefully prepared to incorporate the worst case combinations of these variables so that qualification will not be limited to an average configuration but will embrace the full gamut of manufacturing perturbations.

The above procedure can become very expensive, and thus its implementation must be based on cost effectiveness considerations with due regard to function criticality and the long term economics that result from proven reliability.

Since bearings constitute the most critical failure mechanism, the above recommended routine probably can be limited to the bearings and construction features which influence bearing operation. In some instances, other facets of motor design may need the same treatment i.e., close air gaps, gearing misalignments etc.; however, much of this should be avoidable by stringent worst case analysis.

a. Accelerated Testing - As discussed earlier, the primary wear-out mechanisms of motors involve the bearings, gears and brushes. The latter, however, with short life capability, can be tested in real time.

The fatigue testing of rolling element bearings constitutes a classical approach to accelerated testing as discussed in Section C2a1. However, this technique assumed adequate and infinite lubrication. Therefore it is not appropriate to spacecraft although any bearing used in spacecraft must have adequate fatigue life capability.

Lubricant life must be assessed by accelerated testing. Life is a function of evaporation rate, migration and degradation. Since these are all a function of temperature, then temperature becomes the obvious parameter by which to accelerate these processes.

Certain complications arise as a result of increasing temperature. While increasing evaporation rate, it also decreases viscosity which diminishes film thickness. This changes the operating circumstances which it was the original intent to test.

These complications can be compensated for by utilizing a higher viscosity oil of the same basic formula that has a similar evaporation rate and viscosity at the higher (accelered) temperature. In the case of mineral oils, this can be accomplished by molecular distillation of a specific cut; and, in the case of synthetic oils, by blending. By this means the film thickness should remain unaffected (assuming the visco/pressure characteristics of the test oil is unchanged from the original oil. See Reference 12 which recommends this approach.

2. Life Tests

Until accelerated testing is established as a reliable/predictable process, long life testing of motors must be predicted from short term (practical) tests. The wear rate of bearings and gears are monitored throughout the initial wear-in period until wear rate stabilizes. Life must then be projected by comparing this wear-rate data with known life statistics of similar mechanisms. Likewise the measured loss of lubricant compared with total oil reservoir capacity must signify adequate oil reserves--since the mechanisms of oil transfer from the reservoir is still inadequately known.

One of the most important facets of life testing is that taken up by vibration testing--during which the motor is probably not operating. This is particularly true of motors and bearings used in shuttle type programs where vibration testing may involve hours of testing in each axis. Compared to the degradation which such vibtation testing could incure, the wear resulting from a normal (operating) life test, could be relatively insignificant. (Bearings may have to be sized, primarily to resist the vibration environment.)

It follows from this argument that life and vibration testing should be alternated in a manner representative of an actual shuttle program in order to acquire a realistic degradation pattern.

3. Screening

The metallic components of motor bearings should be individually inspected for accuracy, surface finish and blemishes. The non-metallic retainer, in addition to the preceding shall be inspected for complete freedom from burrs. This data shall constitute a permanent record together with test data of the assembled bearing giving breakaway torque data and electronic noise analysis data.

The complete motor (or motorized mechanism, in the case of frameless motors) shall be subject to an extended run-in (acceptance) test. The duration is usually of several hours (dependent on service life and duty cycle) during which load and the most severe combination of ambient temperature, ambient pressure and heat sink capacity shall be simulated. This should be followed by a nominal room temperature test. Motor performance, including winding resistance and bearing or bearing housing temperature, shall be continuously monitored and shall constitute a permanent part of the record. The record shall also cover any prior running-in incurred as part of the manufacturing process. The record should evidence no perturbation from normality. By this means, failure trends can be recognized and infant mortality detected and eliminated.

In the case of brush type motors, particularly in the high speed variety, the extent of the run-in test may consume valuable brush life. In such instances, it may be advisable to install a new set of pre-contoured brushes followed by a final performance check, prior to delivery.

4. Failure Mode Detection

a. Brush Wear - In the case of brush type motors, the brush springs can be instrumented with a strain gage calibrated to spring deflection. This has been successfully accomplished on despin motors on classified satellites. The brush wear has been accurately recorded. This method can be used in conjunction with either telemetry on satellites or alternately, test terminals could be incorporated on shuttle type vehicles for ground checking. (It is realized that brushless motors avoid this failure mode, nevertheless slow speed applications may still arise where the simplicity of brushes outweighs the complexity of electronic commutating.)

b. Bearing Degradation - Whether from contamination or wear, or lack of lubricant or structural deformation, bearing degradation will cause excess torque or motor load, demanding more power or current. Hence, monitoring of motor performance can result in a direct indication of a failure mode. However, it might not stem from a bearing failure in the motor; it could be due to excess friction anywhere in the driven mechanism. However, for simple applications such as reaction wheels, gyros and mirror or chopper actuation in optical instruments, this is not the case; current monitoring should be very indicative of incipient failure.

c. Infant Mortality - Infant mortality of motors due to bearing problems should be eliminated by the stringent inspection of bearing assemblies and bearing components. Ref. Section B.2.2)g.

d. Mechanical Signature Analysis - The noise signatures of motors using vibration spectrum analysis could be developed to a practical art on shuttle type vehicles. Obviously a signature degradation pattern vs operating time would have to be developed from actual or accelerated life test programs wherein the motor installation is closely simulated. Reference 13 discusses the necessary type of equipment and methods of defect detection.

E. PROCESS CONTROL REQUIREMENTS

The recommended process control requirements have been presented in previous text. Reference was made to the following reproductions of a U.S. Government memorandum on *Procedures for Cleaning and Vacuum Impregnating Bearings* and a *Procedure for Vacuum Impregnation of Porous Bearing Ball Separators*, MRDB-002M.

1. Memorandum by Babecki

UNITED STATES GOVERNMENT

Memorandum

TO : C. Thienel, Code 450
Nimbus Project Office

DATE: August 28, 1970

FROM : A. J. Babecki, Code 764
Materials R&D Branch

SUBJECT: Procedures for Cleaning and Vacuum Impregnating Bearings

The attached two procedures were prepared for use on Numbus E experiments and systems that are to be lubricated with oil or oil and grease. In either case, the ball separators should be made of reinforced laminated phenolic or sintered nylon as first and second choice materials in order to obtain maximum life of operation.

For bearings to be operated only a relatively few cycles, the hardened steel crown ball separators or the soft steel ribbon retainers, in that order, would be acceptable. The attached procedure for cleaning the bearings would still be applicable.

Recent problems with non-wettable surfaces on ball bearings, as received from the manufacturer, have caused concern that the lubrication of such bearings is seriously jeopardized and have prompted the issuance of these written procedures.

If the bearing manufacturer is to perform the lubrication of the bearings, including the vacuum impregnation of the separators, the written procedures of said manufacturer should be obtained for review.

A J Babecki

A. J. Babecki

cc: H. Frankel (2)
W. Cherry
A. Eubanks
A. Babecki
A. Fisher
T. Sciacca
R. Bourdeau
R. Ziemer
H. LaGow
M. Moseson
S. Weiland
J. Lovelace
H. Neumann
W. Bailey

PROCEDURE FOR CHLOROFORM CLEANING OF NEW BALL BEARINGS

August 26, 1970 #MRDB-001M

This cleaning process is recommended for cleaning ball bearings as received from the bearing vendor for two reasons:

1. To remove the preservative oil that was applied by the manufacturer and to help ensure clean wettable surfaces for the operating lubricant to be applied.

2. To increase the porosity of laminated phenolic ball separators that are to be vacuum impregnated with the operating oil lubricant.

This cleaning procedure may be performed on ball bearings in either the assembled or disassembled states. If assembled, the bearings should have the shields (if equipped with them) removed. However, the shield should be subjected to the same cleaning process.

The chloroform to be used should be fresh, unused bottled grade.* The glassware and other containers and handling tools should be clean and should be rinsed in the clean chloroform before use. After removal of the shields, the bearings should be handled with clean tweezers, tongs, or other non-porous tools.

The bearing parts should be totally immersed in the chloroform bath with at least one inch of coverage and gently stirred or agitated for 5 minutes to remove the bulk of the preservative oil and/or contamination. The bearing parts should be removed to another clean container and the process repeated; the first bath and subsequent baths should be discarded after use.

The bearing parts should be transferred to a fresh chloroform bath which is then heated to 125-150^{o}F on a steam table or hot plate. With gentle agitation, this cleansing should continue for 15 minutes after which the parts are transferred to a fresh bath, and the hot 15-minute cleansing repeated.

Following the above step, the parts should be withdrawn one at a time and jet rinsed with clean room temperature chloroform from a laboratory wash bottle and placed in a clean container for drying in a non-circulating oven at 150-200^{o}F for at least one hour.

---

*Certified ACS Spectranalyzed chloroform

The parts should then be packaged in chloroform - cleaned aluminum foil or glass petri dishes for storage or transfer to the lubrication area. If the lubrication is to be delayed more than six hours, the packaged bearing parts should be stored in a tight desicoated container or under vacuum.

A. J. Babecki

PROCEDURE FOR VACUUM IMPREGNATION OF POROUS BEARING BALL SEPARATORS

August 26, 1970 #MRDB-002M

This impregnation process may be performed on any porous bearing ball separator material - phenolic, sintered nylon, sintered metal. Prior to the impregnation, the separators should be thoroughly cleaned, preferably in hot (125-150°F) chloroform as outlined in procedure MRDB-001M.

The vacuum impregnation may be performed on individual ball separators or on assembled bearings with shields removed. More than one separator or bearing may be impregnated in the same oil bath.

The oil to be impregnated into the separators is poured into a chloroform - cleaned tall glass beaker or vial to a depth sufficient to cover the bearing parts with approximately one-half inch over coverage. This oil bath is then placed in a vacuum oven and heated to 125-150°F and deaerated at a vacuum level of 1×10^{-3} mm (1 micron) until all bubbling ceases. The vacuum may have to be pulled gradually to prevent frothing of the oil over the edge of the container.

After the vacuum is broken to atmospheric pressure with air, the bearing parts are gently immersed into the oil bath with minumum agitation, and the oil bath is placed again into the vacuum oven.

The oil bath is again heated to the temperature of impregnation (125-150°F) to reduce the viscosity of the oil, and the vacuum of 1×10^{-3} mm is again slowly drawn to prevent frothing over the container edge.

The vacuum is maintained until all bubbling ceases. This period may last several hours until the fine bubbling has stopped. Overnite exposure to the vacuum at the elevated temperature is a convenient method of ensuring exhaustion of all entrapped air from the separators.

After all bubbling has ceased, the vacuum is broken back to atmospheric pressure, the heat is turned off, and the bearing parts are allowed to cool to room temperature while submerged in the oil bath. During this cool-down period, the density of the oil increases and a greater weight of oil is impregnated.

After cool down, the bearing parts are removed and may be packaged in clean containers as is, or the excess oil may be removed (if desired) by any of several methods, e.g., centrifuging, a blast of clean gas (N_2), or blotting or wiping with lint-free wipers.

If knowledge is desired of the quantity of oil introduced into the bearing, the individual bearings or separators should be weighed to 0.1 mgms. before the impregnation and after removal of the external excess oil following the impregnation.

A. J. Babecki

F. REFERENCES AND BIBLIOGRAPHY

1. References

1. R. O. Froling: Engineering Reports for Motor Driven Switch--AGE Degradation Study and Failure Analysis, March 1971, August 1971, and September 1971, Martin Marietta Corporation, Denver, Colorado.

2. J. C. Tyler, R. A. Burton and P. M. Ku: "Contact Fatigue Under Oscillatory Normal Load." *ASLE Transation 6 255-269*, 1963.

3. R. A. Burton, J. C. Tyler and P. M. Ku: "Thermal Effects in Contact Fatigue Under Oscillatory Normal Load." *ASME Journal of Basic Engineering*, March 1965.

4. Tedric A. Harris: *Rolling Bearing Analysis*. John Wiley & Sons, Inc., New York.

5. Knudsen, M.: *Kinetic Theory of Gasses*. Third Edition, Methuen and Company, Ltd., London, 1950.

6. Lester, S. P.: *Labyrinth Sealing of Bearings for Vacuum Environment*. Report MT 13285, Bendix Corporation, Teterboro, New Jersey, August 1966.

7. D. B. Hamilton and J. S. Ogden: *The Evaluation of Various Lubricant Fluids in Vacuum*. Battell-Columbus Laboratories, Columbus, Ohio.

8. J. T. McCabe and T. Y. Chu: *Grease Lubricated Spiral Groove Gyro Bearings*. FIRL, Final Report F-C2028-1, Contract NAS8-20741.

9. H. Mura: *Stability Characteristics of Gyroscope with Hydrodynamically Grooved Rotor Bearings*. NASA Report TWX-53744, Marshall Space Flight Center.

10. Herbert B. Singer: *The Effect of TCP Treatment on the Low-Speed Performance of Ball Bearings*. Report No. E2317, MIT Instrument Laboratory, 1968.

11. Albert P. Freeman, Steven Allen and Herbert B. Singer: *Ball Bearing Surface Chemistry*. Report No. E2288, the Instrument Laboratory of Massachusetts Institute of Technology, September, 1968.

12. C. R. Meeks, R. I. Christy and A. C. Cunningham: "Accelerated Vacuum Testing of Long Life Ball Bearings and Sliprings." *5th Aerospace Mechanisms Symposium.*

13. A. S. Babkins and J. J. Anderson: *Mechanical Signature Analysis of Ball Bearings by Real Time Signature Analysis.* Paper presented at the Annual Meeting of the IES, New York, May 1972.

2. Bibliography

1. D. E. Lyons: *Lubrication Handbook for Use in Space Industry.* A&TS-PR-M, NASA/MSFC.

2. *Life Adjustment Factors for Ball and Roller Bearings.* ASME, September 1971.

3. *American Standard Method of Evaluating Load Ratings for Ball and Roller Bearings.* ASA Standard 133.11.1959.

4. Tibor Tallian: *Weibull Distribution of Rolling Contact Fatigue Life and Deviations Therefrom.* Asle Transactions, Volume 5, Number 1, April 1962.

5. J. I. McCool: *Load Ratings and Fatigue Life Prediction for Ball and Roller Bearings.* The American Society of Mechanical Engineers Lubrication Symposium, San Franciso, California, June 16-19, 1969.

6. T. E. Tallian: *An Engineering Model of Spalling Fatigue Failure in Rolling Contact - III. Engineering Discussion and Illustrative Examples.* ASME/ASLE Lubrication Conference, Cincinnati, Ohio, October 13-15, 1970.

7. T. E. Tallian and J. I. McCool: *An Engineering Model of Spalling Fatigue Failure in Rolling Contact - II. The Surface Model.* ASME/ASLE Lubrication Conference, Cincinnati, Ohio, October 13-15, 1970.

8. Y. P. Chiu, T. E. Tallian and J. I. McCool: *An Engineering Model of Spalling Fatigue Failure in Rolling Contact - I. The Subsurface Model.* ASME/ASLE Lubrication Conference, Cincinnati, Ohio, October 13-15, 1970.

9. J. Lieblein and M. Zelen: *Statistical Investigation of the Fatigue Life of Deep-Groove Ball Bearings.* Journal of Research of the National Bureau of Standards, Vol. 57, No. 5, November 1956, Research Paper 2719.

10. J. W. Kannel, J. A. Walowit, J. C. Bell and C. M. Allen: *The Determination of Stresses in Rolling-Contact Elements.* Paper No. 66-Lub-16, ASME-ASLE Lubrication Conference, Minneapolis, Minn., October 18-20, 1966.

11. R. K. Kepple and R. L. Mattson: *Rolling Element Fatigue and Macroresidual Stress.* The American Society of Mechanical Engineers Lubrication Symposium, San Francisco, California, June 16-19, 1969.

12. F. G. Rounds: *Lubricant and Ball Steel Effects on Fatigue Life.* ASME-ASLE Lubrication Conference, Cincinnati, Ohio, October 12-15, 1970.

13. Robert L. Johnson and Lawrence P. Ludwig: *Lubrication, Friction, and Wear in Aricraft.* Lewis Research Center, Cleveland, Ohio, NASA TM X-67872, March 1971.

14. Marshall B. Peterson: *Literature Search and Critical Review of Mechanical Failure Technology.* Mechanical Technology, Incorporated, July 1970.

15. Robert P. Pardee: *The Effect of Humidity on Low-Load Frictional Properties of a Bonded Solid Film Lubricant.* ASME/ASLE Lubrication Conference, Pittsburgh, Pennsylvania, October 5-7, 1971.

16. F. J. Westlake and A. Cameron: *Optical Elastohydrodynamic Fluid Testing.* ASME/ASLE Lubrication Conference, Pittsburgh, Pennsylvania, October 5-7, 1971.

17. Talivaldis Spalvins and Donald H. Buckley: *Vapor-Deposited Thin Gold Films as Lubricants in Vacuum (10^{-11} mm Hg).* Lewis Research Center, Cleveland, Ohio, NASA TN D-3040, October 1965.

18. Marshall W. Dietrich, Dennis P. Townsend and Erwin V. Zaretsky: *Rolling-Element Lubrication with Fluorinated Polyether at Cryogenic Temperatures (160° to 410° R).* Lewis Research Center, Cleveland, Ohio, November 1969, NASA TN D-5566.

19. E. J. Devine, Harold E. Evans and William A. Leasure: *Instrument Bearing Performance in Space and Laboratory Vacuum*, NASA TN D-6035. Goddard Space Flight Center, Greenbelt, Md., October 1970.

20. L. Dale Smith, Dean C. Glenn and Herbert W. Scibbe: *Evaluation of Impregnated Lubricants in Ball-Bearing Retainers at 10^{-6} Torr*, NASA TN D-3259. Lewis Research Center, Cleveland, Ohio, February 1966.

21. A. Kampinsky: *Evolution of a Spacecraft Antenna System*, NASA TN D-6483. Goddard Space Flight Center, Greenbelt, Md., September 1971.

22. N. I. Silversher and S. P. Drake, Jr.: *Final Report, Lubrication Evaluation*, MRI 503.02. Lockheed Missiles and Space Company, Sunnyvale, California, April 1968.

23. Erich W. Manteuffel, *et al.*: *Brushless DC Torque Motor Development*, AD 707 514. General Electric Company, Binghamton, New York, January 1970.

24. Donald H. Buckley: *The Behavior of Lubrication System Components in a Vacuum Environment*. NASA Lewis Research Center, Cleveland, Ohio, presented at 2nd Aerospace Mechanisms Analysis Symposium, 1967.

25. C. L. Harris, J. E. Read, J. B. Thompson and C. I. Wilson: *Lubrication in Space Vacuum - Part 2. Life Test Evaluation of Solid Film Lubricated Ball Bearings at 10^{-10} Torr*. Lubrication Engineering, February 1968.

26. C. L. Harris, J. E. Read, J. B. Thompson and C. I. Wilson: *Lubrication in Space Vacuum - Part 1. Testing Techniques for Ball Bearings*. Lubrication Engineering, 1968.

27. C. L. Harris, J. E. Read and J. B. Thompson: *Lubrication in Space Vacuum - Part 3. Life Test Evaluation of Ball Bearings Lubricated with Oils and Grease*. Lubrication Engineering, April 1968.

28. Francis J. Clauss: *Lubrication as Part of Total Design*. Presented at 2nd Aerospace Mechanisms Symposium 1967.

29. H. I. Silversher: *Controlled-Leakage Sealing of Bearings for Fluid Lubrication in a Space Vacuum Environment.* Presented at 3rd Aerospace Mechanism Symposium, 1968.

30. D. L. Kirkpatrick and W. C. Young: *Evaluation of Dry Lubricants and Bearings for Spacecraft Applications.* Presented at 3rd Aerospace Mechanism Symposium, 1968.

31. R. L. Johnson and D. H. Buckley: *Lubricants and Mechanical Components of Lubrication Systems for Space Environment.* ASLE 19th Annual Meeting, Chicago, Illinois, May 1964.

32. H. I. Silversher: *Materials for Lubrication Applications.* Chapter 9, Space Materials Handbook, Third Edition, 1968, Air Force Mateials Lab Report No. AFML-7R-68-205.

33. C. L. Harris: *Lubricants in Space, Part 1 Bearings.* Mechanical Engineering, April 1970.

34. D. B. Hamilton and J. S. Ogden: *The Evaporation of Various Lubricant Fluids in Vacuum.* Battelle-Columbus Laboratories.

35. C. R. Meeks, R. I. Christy and A. C. Cunningham: *Accelerated Vacuum Testing of Long Life Ball Bearings and Sliprings.* Transactions of the 5th Aerospace Mechanisms Symposium.

36. J. W. Kannel and J. C. Bell: *A Method for Estimation of Temperatures in Lubricated Rolling-Sliding Gear on Bearing Elastohydrodynamic Contacts.* Institution of Mechanical Engineers Elastohydrodynamic Lubrication Symposium, London, 1972.

37. S. W. Poon: *An Experimental Study of Grease in Elastohydrodynamic Lubrication.* Journal of Lubrication Technology, January 1972.

38. H. S. Cheng: *Isothermal Elastohydrodynamic Theory for the Full Range of Pressure-Viscosity Coefficient.* Journal of Lubrication Technology, January 1972.

39. P. E. Fowles: *Extension of the Elastohydrodynamic Theory of Individual Asperity-Asperity Collisions to the Second Half of the Collision.* Journal of Lubricaton Technology, April 1971.

40. J. W. Kannel and J. A. Walowit: *Simplified Analysis for Tractions Between Rolling-Sliding Elastohydrodynamic Contacts*, Battelle Memorial Institute, Columbus, Ohio. Journal of Lubrication Technology, January 1971.

41. J. C. Skurka: *Elastohydrodynamic Lubrication of Roller Bearings*. Journal of Lubrication Technology, April 1970.

42. J. K. Kannel and J. A. Walowit: *Simplified Analysis for Tractions Between Rolling-Sliding Elastohydrodynamic Contacts*, Battelle Memorial Institute, Columbus, Ohio. Journal of Lubrication Technology, January 1971.

43. J. W. Kannel and J. C. Bell: *Interpretations of the Thickness of Lubricant Films in Rolling Contact. 1. Examination of Measurements Obtained by X-Rays*, Battelle Memorial Institute, Columbus, Ohio. Journal of Lubrication Technology 1971.

44. J. C. Bell and J. W. Kannel: *Interpretations of the Thickness of Lubricant Films of Rolling Contact. 2. Influence of Possible Rheological Factors*. Journal of Lubrication Technology, 1971.

45. C. A. Foord, L. D. Wedeven, F. J. Westlake and A. Cameron: *Optical Elastohydrodynamics*. Volume 184, Part 1, Number 28, The Institution of Mechanical Engineers, Proceedings, London, England, 1969-70.

46. V. G. FitzSimmons, *et al.*: *A new Approach to Lubricating Ball Bearings*, AD 627 345. U.S. Naval Research Laboratory, Washington, D.C., December 1965.

47. J. C. Bell and J. W. Kannel: *Simulation of Ball-Bearing Lubrication With a Rolling-Disk Apparatus*, Columbus Laboratories, Columbus, Ohio. The American Society of Mechanical Engineers Spring Lubrication Synposium, San Francisco, California, June 17-19, 1969.

48. A. R. Leveilk, R. R. Delletier and R. J. Giannott: *Aerospace Bearing-Analytical Investigation by New Departure-Hyatt Bearings*. For Air Force Flight Dynamics Laboratory, WPAFB, Ohio, Report No. AFFDL-TR-66-134, January 1967.

49. J. B. Pearcé, K. A. Gause, E. F. Mackey, K. K. Kelly, W. G. Fastir and C. A. Barth: *Mariner 6 and 7 Ultraviolet Spectrometers Applied Optics*, Vol. 10, April 1971.

50. Philip A. Studer: *Development of a Sealed Brushless DC Motor*, NASA TN D-2819. Goddard Space Flight Center, Greenbelt, Md., May 1965.

51. *Design, Development, and Fabrication of a Sealed, Brushless DC Motor*, NASA CR-231. Contract No. NAS5-3582 by Sperry Farragut Company, Briston, Tennessee, May 1965.

52. G. H. Fountain and B. M. Phillips: *SAS-A Momentum Wheel*. The Johns Hopkins University, Applied Physics Laboratory, APL/JHU, CP 012, February 1972.

III. ACCELEROMETERS

by R. Opper

III. ACCELEROMETERS

A. INTRODUCTION

This chapter sets forth factors concerned with establishing a known performance potential for linear accelerometers over a ten year period. The accelerometers studied are those applicable to manned spacecraft used in navigation, stability and control functions.

This chapter is concerned with the failure mechanisms, test and periodic calibration methodology, manufacturing processes and other controls to achieve the life objectives. This chapter concentrates upon the sensing instrument itself, and not the driving electronics and other periphery elements because the scope is limited to three principle types of accelerometers: the pendulous, the unbalance gyro type (PIGA or pendulous integrating gyro accelerometer)* and the vibrating string accelerometer. Section G describes these accelerometers for those desiring more orientation information. The vibrating string accelerometer has been phased out of operational use. There are many other concepts. The technology may be regarded to be in a state of flux; however, with reliance upon experience for the long life data, the aforementioned limitation is made. Figure 1 sets forth the types of accelerometers considered in this report. Solid state accelerometers (piezoelectric) were not considered in any detail in this report principally because sensitivity and threshold performance achieved with this type sensor does not permit its use as a guidance and control instrument. A significant breakthrough with this instrument could permit its use, which would be desirable because of its simplicity and reliability.

The information provided is for a class of accelerometer instruments. The information is independent of specific performance specifications that are highly dependent upon the particular mission of the space vehicle and the accuracy and control aspects of those missions. Since the instruments selected are typical of general classes, they should represent and incompass the type that would be ultimately applied in the next few years.

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*The unbalanced gyro accelerometer is commonly referred to as the Pendulous Integrating Gyro Accelerometer or PIGA. The term PIGA will be used in this report.

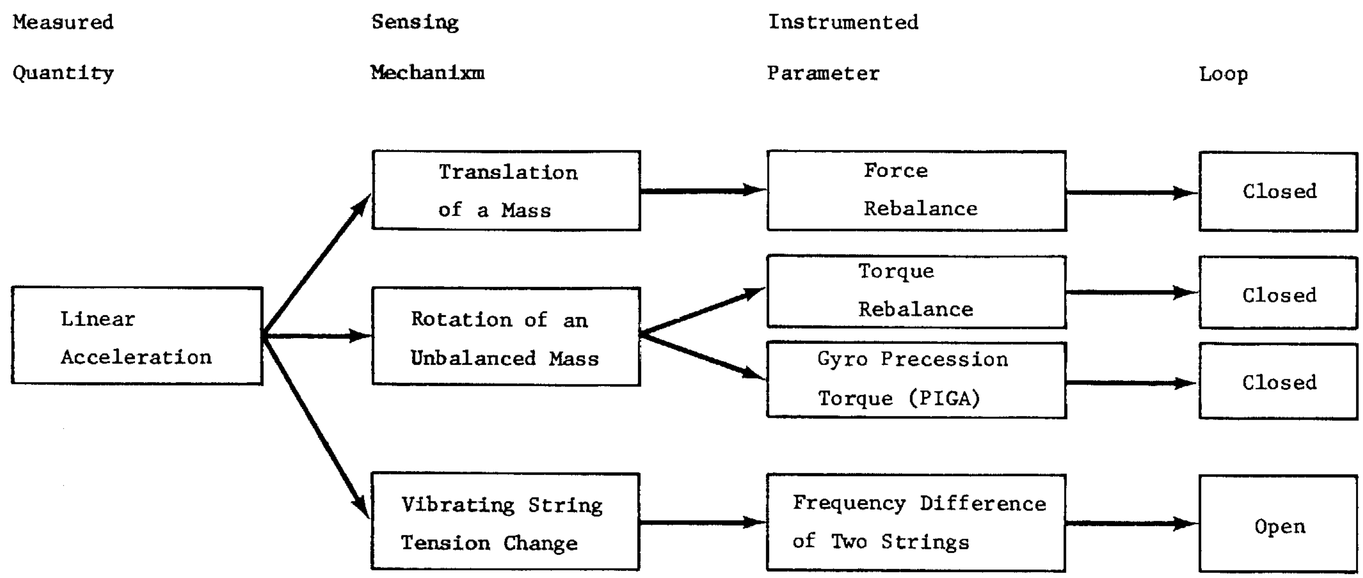

Figure 1 Types of Considered Accelerometers

B. GUIDELINES FOR LONG-LIFE ASSURANCE

Of the three types of accelerometers considered, only the force feedback pendulous (or proof mass) type accelerometer has the potential for performance over the extended time. With the incorporation of proper design practices, manufacturing and test processes, a ten year life, operating or non-operating, is obtainable. Bias and scale factor must be measured periodically because of instability. Some recalibration will be required. Trends in scale factor changes can be shown from the periodic test data that are accountable by permanent magnet aging. The only significant item that ages is the permanent magnet material which follows a predictable time relationship.

1. Design Guidelines

1) Use pendulous or proof mass accelerometer with force feedback. It has no rubbing or mechanical friction in the design. The technology has been developed to a state where long-life is inherent in the design.

2) Use a fluid filled accelerometer; they can withstand the environmental effects of pyro shock and vibration.

3) Select a basic accelerometer capable of greater performance than is required. Performance margin will result in fewer specification requirements because the scale factor degrades with time and the bias instability has long term trends.

4) Employ a magnet with a higher shape factor (L/D) to increase the remanence stability. Also, the higher the coercive force the more stable will be the remanence. Highly crystal oriented ALNICO-5 is significantly more stable than the normal random oriented materials.

5) Use pure properly treated metals to enhance long-life performance by reducing microcreep.

2. Process Control Guidelines

1) Cleanliness of the inert fluid is mandatory to prevent contamination. There have been a number of instances of contamination failure in space vehicles. Furthermore, cleanliness of all instrument material is generally mandatory.

2) Precondition material to reduce creep and to enhance magnetic stability. These instabilities are greater at the outset of the performance life. Employ artificial magnet remanence reproduction by temperature cycling or small ac fields. The remanence of a magnet will be stabilized.

3. Test Guidelines

1) Periodic tests, depending upon specific requirements for the accelerometer, will be required to determine trends and for recalibration for changes that occur in bias and scale factor.

2) There is no known accelerated testing technique on complete accelerometer assemblies.

3) Screening processes or wear-in tests are sometimes employed for assembled accelerometers (beyond the standard checkout and acceptance testing). Since accelerometers are semi-passive devices, such tests are not universally employed. Wear-in is most applicable to the PIGA accelerometer because this type is subject to wear when it contains ball bearings.

4. Special Considerations

1) Use designs with a proven history and experience because the potential problems of unknown and new designs is a major risk over the extended time period.

2) Either exercise or change the "storage" position periodically to prevent adjustment change of the critical axis bias. Long storage in the same position can affect accuracy.

C. LIFE LIMITING PROBLEMS AND SOLUTIONS

1. Failure Mechanism Analysis

Table 1 presents the results of the failure mechanism analysis. This table summarizes some of the failure mechanisms prevalent in accelerometers. The scope is limited to the accelerometer instrument itself and not the driving electronics. The driving electronics and the pickoff electronics are a combination of parts that are a subject of a companion report (Gyroscopes and Bearings, Chapter IV.) Permanent magnet aging, fluid contamination, bias change and bellows leaks are discussed in the following subsections.

Accelerometers of the pendulous type have no rubbing parts and no friction; therefore, they are not subject to a wearout in the usual sense. The PIGA are subject to the same life limiting factors as a gyro, plus the additional torque motor bearings and slip rings.

The vibrating string accelerometer is subject to microcreep in the string material. Generally, in practice, where greater accuracy is required, periodic calibration of the accelerometer is required to maintain the accelerometer within a tolerance band.

The original manufacturing and design process, normally applied to accelerometers, are extremely exacting because of the performance stability requirements. The design and manufacturing processes required for high stability are in most instances the same as required for long life--you get long life for nothing when stability is achieved. Flexures, when used, are carefully chosen for high modulus of elasticity; the materials are chosen for minimum creep and maximum stability; fluids are chosen to be inert (silicon or fluoro compounds); proven encapsulates are used, and the rubber seals, if used, are long life silicon types.

a. *Permanent Magnets* - The most prevalent aging effect is the aging in the permanent magnets. Figure 2 illustrates the exponential decay of magnetic strength. Preconditioning takes most of the aging from the permanent magnets. The permanent magnet stability, as related to long life and applied to accelerometers, are influenced by (Reference 1):

Table 1 Failure Mechanism Analyses - Accelerometers

Failure Mode	Probability of Occurrence	Effects	Mechanism	Detection	How to Eliminate/ Minimize Failure
a) Permanent Magnet Aging	Nearly 100% for ALNICO	Performance change & Scale Factor change yielding system errors	Aging	Periodic ground test. Log change relationship is predictable	Select accelerometers with greater performance margin.
b) Fluid Contamination (Except vibrating string)	Slight	Erratic output. Particles accumulate in critical tolerance regions	Contamination	Periodic ground test	Better process control
c) Bias Change (creep, micro-creep)	Depends on spec tolerance could be 100%	Output with no input--system error	Aging	Periodic ground test will indicate trend	Select accelerometers with greater performance margin
d) Bellows Leak	Slight	Gas in fluid	Cycling	Periodic ground test will show instability	Initial design
e) PIGA Wheel Failure	Nearly 100%	No output from gyro--bearing failure	Wearout	For oil lubricant, periodic test will indicate a recondition of instability. For gas lubricant, detection method does not exist. NOTE: Gas is potentially longer life	

*NOTE: See Chapter IV for unbalanced gyro failure mechanism analysis. Unbalanced gyros are subject to the above, plus bearing wearout.

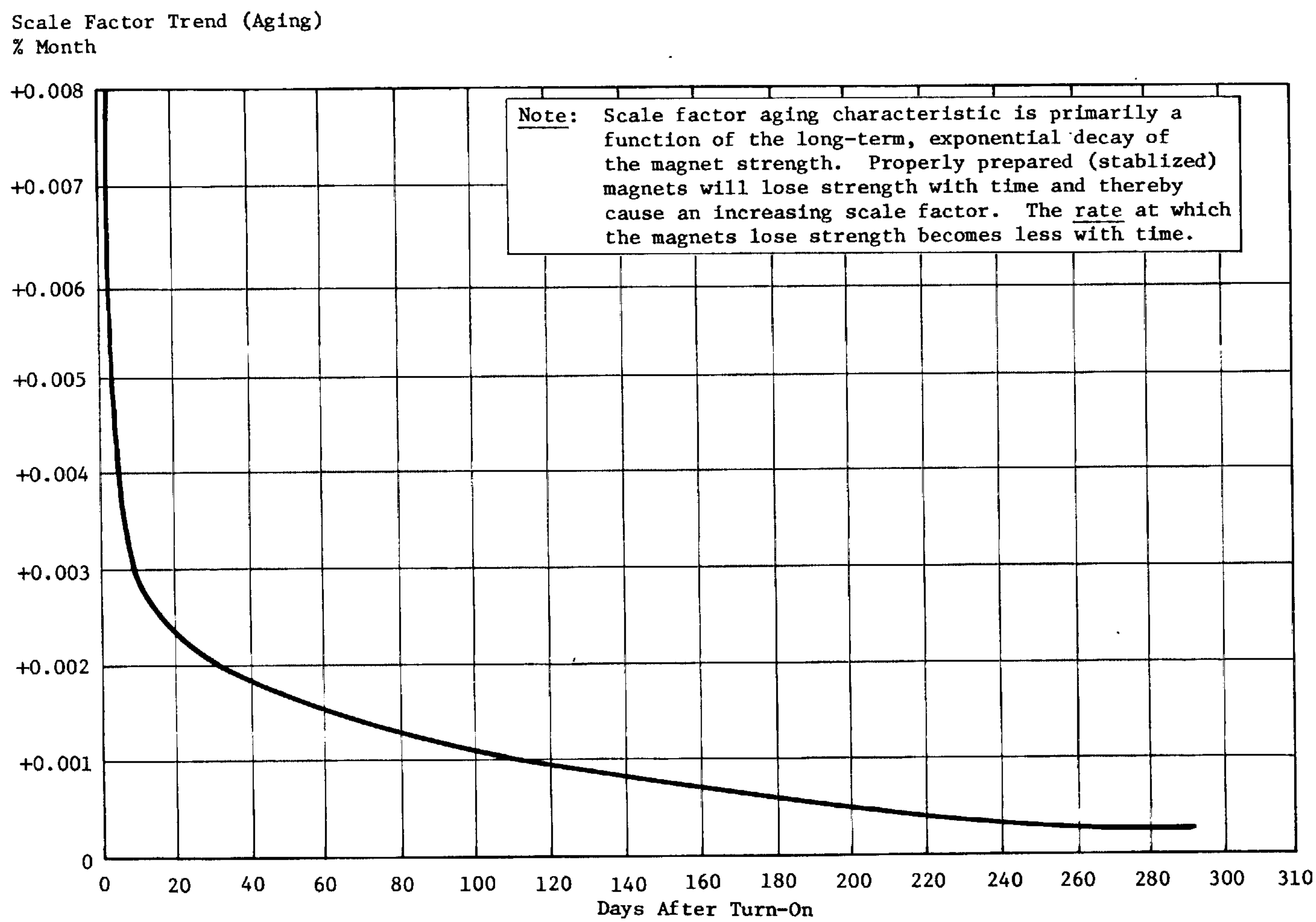

Figure 2 Typical Long Term Scale Factor Aging Characteristics

1) Temperature Effects;

2) Mechanical Stresses;

3) Relaxation Effects;

4) Radiation.

All permanent magnets are subject to disturbing influences to some degree. With careful processing, the magnitude of influence can be estimated. The general procedure is to stabilize the magnets at a condition somewhat in excess of the condition of actual service to achieve the condition of maximum stability.

Figure 3 is data on the scale factor of Kearfott Singer Model 2401 accelerometer used in Apollo IX (Ref. 2). The accelerometers were used in a Hamilton Standard (UAC) strapped down system. They are directly related to the predictable log characteristics of the permanence of permanent magnets.

1) *Temperature Effects* - The designer of accelerometers have a wide range of permanent magnetic materials to choose from. The most widely used material is ALNICO-5. Table 2 is related to ALNICO-5. It is common practice for manufacturers to stabilize the magnets by temperature cycling.

Table 2 Temperature Effects on Permanent Magnets

Dimension Ratio L/D	% Irreversible Loss at Room Temperature after Exposure to		Reversible Temperature Coefficient, % Remamanence Change per °C
	-190°C	-60°C	
8.00	0	0	-0.022
5.36	4.6	1.4	-0.012
3.63	9.0	2.5	-0.002
2.72	6.2	3.6	+0.010
1.84	7.9	3.1	+0.016
0.94	8.5	3.4	+0.007

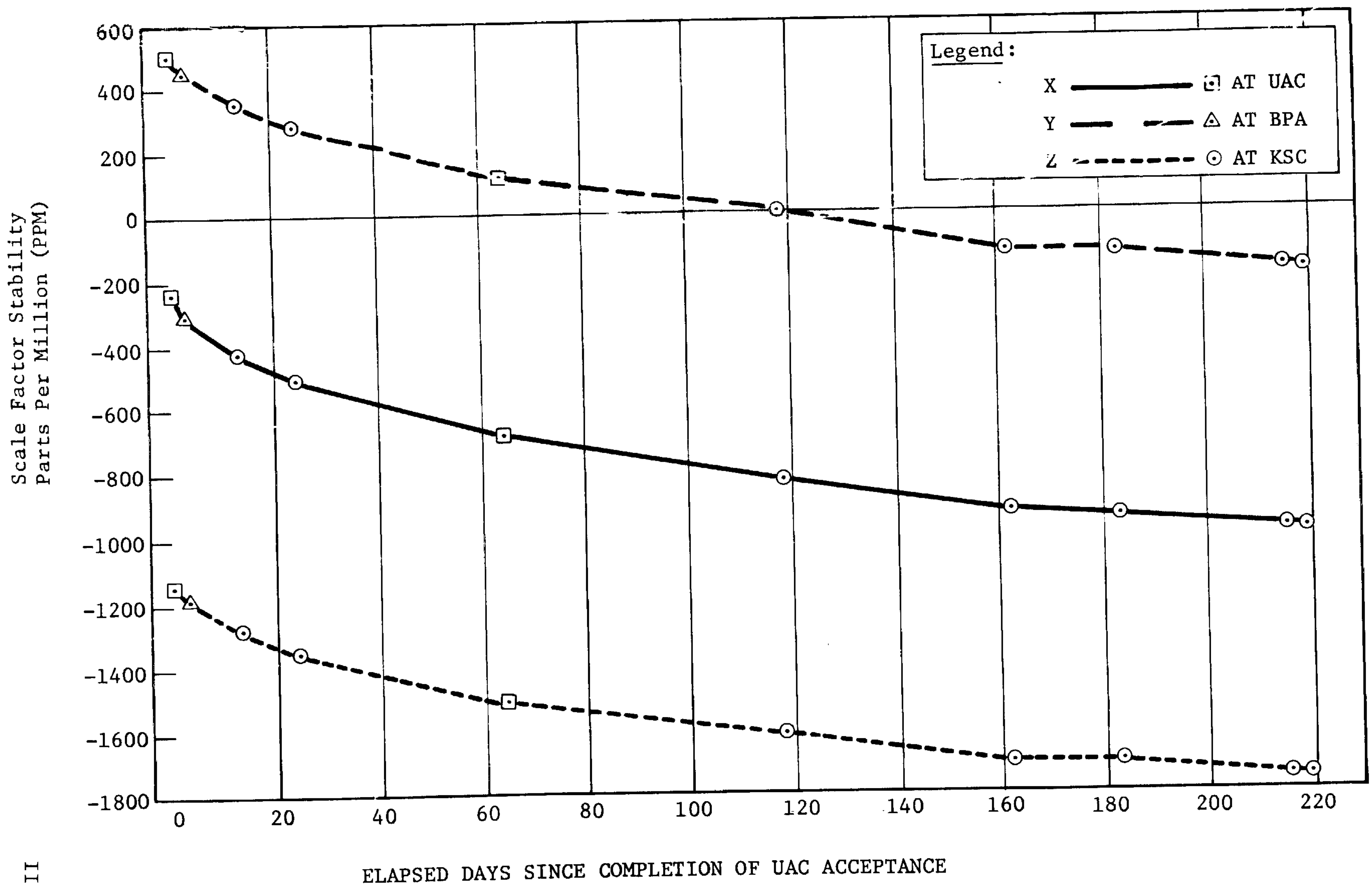

Figure 3 Apollo 9 Accelerometer Data

2) *Mechanical Stress* - Table 3 gives some data on impacts. Manufacturing process normally include preconditioning. These data were taken on 10-inch long bars, 1/2 inch square in cross-section. One impact was equivalent to dropping the magnet a one-meter distance onto a wooden platform, the bar hitting on its end.

Table 3 Mechanical Shock

Percent Remanence Retained by	
No. of Impacts	ALNICO-5
20	100*
50	99.9
100	99.8
200	99.8*
300	99.6
500	99.5
750	99.5*
1000	99.5*

*Indicates region where bars cracked or broke.

Vibration tests can be used to indicate the magnitude of the remanence losses of various magnet materials. Magnet specimens were vibrated with a total displacement of 0.050 inches at frequencies from 10 to 50 cps for a period of 40 minutes. The ALNICO-5 exhibited a 0.32% remanence loss.

3) *Relaxation* - Table 4 presents the summary of experimental results from Kronenberg concerning effect of L/D on magnet stability (remanence). There are several important generalizations possible:

a) The higher the coercive force, the more stable will be the remanence. Highly crystal oriented ALNICO-5 is significantly more stable than the normal random oriented material;

b) The higher the shape ratio, L/D, the more stable its remanence;

c) When the remanence change of a structurally stable and undisturbed magnet is perceptible, the remanence decreases linearly with the logarithm of time for all materials examined;

d) The remanence of a magnet can be stabilized to avoid these magnetization changes by an artificial remanence reproduction by temperature cycling or small ac fields.

Table 4 Summary of Experimental Results from Kronenberg

Material	L/D	Remanence B_d Kilogauss	Stability Relative Remanence at 24°C, 5 Log Cycles (10,000 hr) after Magnetization, %	Measuring Accuracy
Alnico-5	8.0+	1.4	99.95	±0.01
(Long)	6.7	12.1	99.89	±0.02
	5.8	11.9	99.81	±0.02
(Medium)	4.3	10.4	99.23	±0.02
(Short)	3.5	8.2	98.84	±0.04
	3.3	7.6	98.97*	
	2.9	6.7	98.50	±0.05
	2.2	4.8	98.3	±0.07
	2.1	4.1	97.6*	
	1.4	2.6	98.2	±0.1

*Extrapolated 1 to 2 log cycles beyond last measurement.

4) *Radiation* - Permanent magnets have been irradiated for 12 days with 3 x 10^{17} fast neutrons/cc with no detectable change in remanence (Reference 1). Neutron radiation should not be a major problem area.

b. *Fluid Contamination* - There has been sufficient histories of problems with fluid contamination to demonstrate this problem is serious. In the Lunar Orbiter there was evidence of gimbal "hang-up" with contamination being one suspected cause (Ref 3). Gas contamination of Nimbus 3 was one of the suspected causes of excessive drift during flight (Ref 3).

With respect to contamination, the factors influencing life are also those affecting reliability and performance. Contamination cannot be adequately described in a specification. Also the more accurate the accelerometer the more contamination is a problem. To illustrate:

> Assume an accelerometer with a moment of inertia of 0.10 gm. cm^2 and a hinged arm length of 2 cm. A 0.01 cm particle of aluminum logged on the proof mass would cause a scale factor change on the order of 100 parts per million.

Chemical contaminants (such as finger oils, solder fluxes, mold release agents, and moisture absorbed on or into parts) react directly with the fluid in the following manner:

1) Direct Chemical Reaction;

2) Local variations in the flotation fluid properties;

3) Form local droplets with surface tension boundaries.

Other contaminants include burrs, debris from parts, dust from the air, lint from clothing, human hair, human skin particles, bits of solder, and many others. When solvents are used for cleaning parts they leave residue by-products. Freon is one of the better solvents.

Contamination ranks high as a possible source of failure. Documentation is not, by itself, sufficient. Likewise process control, by itself, is not sufficient. Depth of experience is necessary for a complete appreciation of the problems and techniques for contamination elimination.

Tests for compatibility with the fluid may be required to insure material compatibility. Tests for compatibility may be conducted with either the presence of air or in the absence of air. Accelerated testing for compatibility involves raising the temperature, to raise the chemical activity. Typical materials used in contact with the fluid are aluminum, silver, copper, beryllium copper, tungsten, tungsten carbide, brass, 303 stainless, 440 stainless, epoxies, wires insulation, magnetic laminations and solvents.

Proper and extremely careful material choice is important; the manufacturing process to filter and maintain the fluid free of contamination is extremely important. There are generalizations. Inert fluids are generally chosen by the designer. The silicon and fluoro compounded fluids are normally chosen.

c. Bias Change - There are a number of causes for bias change. Careful design and manufacture are the major steps to alleviate the change. Figures 4, 5, and 6 provide some actual data on bias on the Kearfott Singer Type 2401 used on the Apollo IX Hamilton Standard strapdown accelerometers (Ref 2).

One of the design items important to consider is the microcreep in the material (Ref 4). Microcreep is the change in dimension that occurs in a solid-state phase. It is determined as a function of time by the phases present and the degree of the instability. It is also dependent upon the applied stress, the residual stress, and the combination of the two factors. It is also dependent on the response of the internal structure to a force imposed. Therefore microcreep varies with the magnitude and the direction of the load, temperature and the type of material, the preheating or treatment of the material, and the overall time effects of all of these influences.

Generally, the instabilities due to phase changes in metals can be eliminated by using pure metals that have been heat treated or thoroughly brought to a thermodynamic equilibrium. However, pure metals are often either too weak or have other major drawbacks. Techniques of thermal treatments are applied to complete the phase changes as much as possible. The designer of the accelerometer must be aware of this factor in the design; when considered in the initial design it does not constitute a long life factor. *Therefore, pure properly treated metals should be used when possible to enhance long life performance.*

A more major concern is the instability caused by mechanical strain, especially the instability associated with repeated mechanical stresses over a long period of time. In accelerometers the stress arises from acceleration, even though relatively low. It involves a significant amount of material movement over the extended life period. Residual stress caused by the initial manufacture and the temperature are important. The uncertainty of the measure of microcreep further contributes to the uncertainties on whether certain goals are achieved in the initial design.

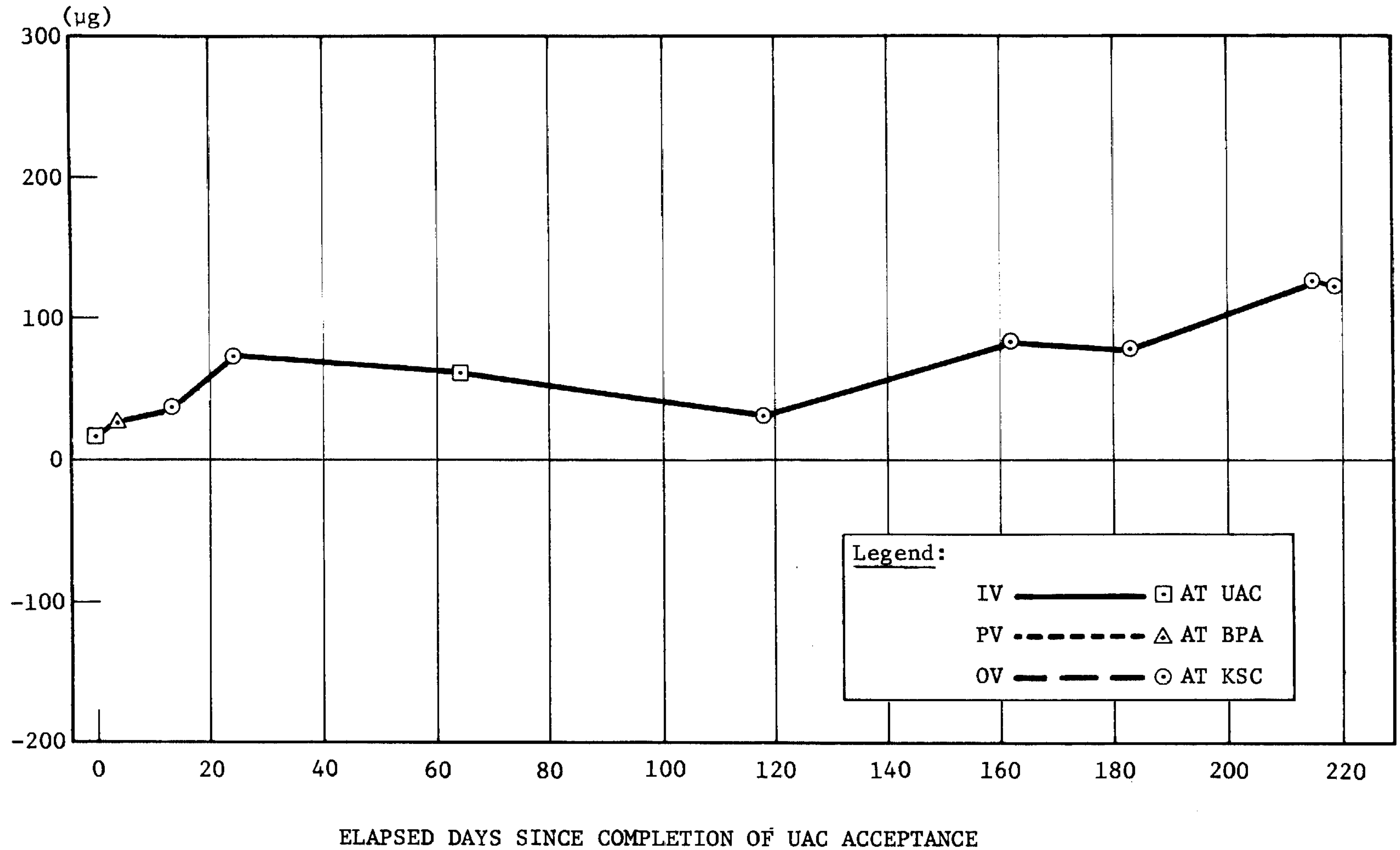

Figure 4 Apollo 9 X-Accelerometer Bias

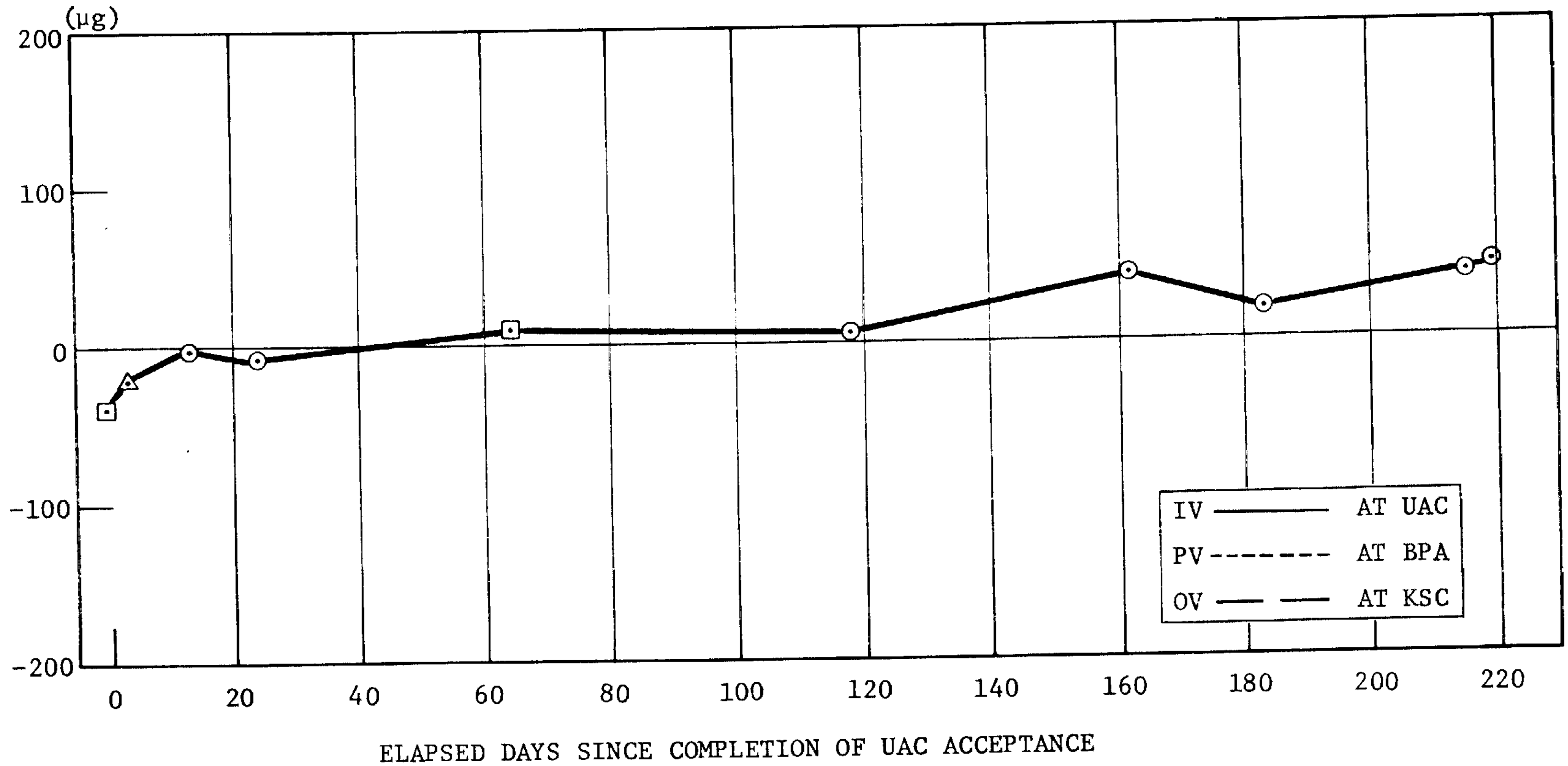

Figure 5 Apollo 9 Y-Accelerometer Bias

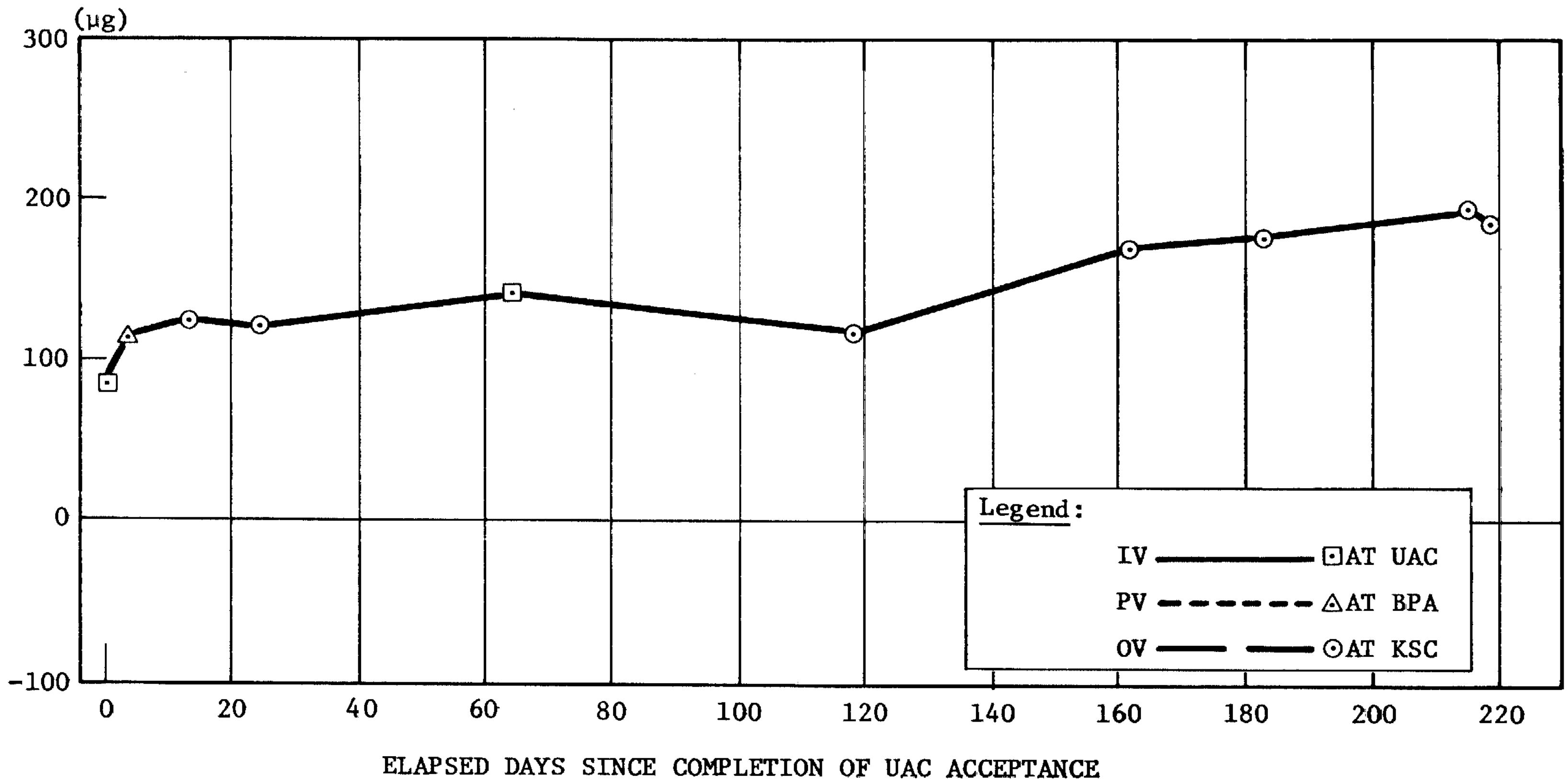

Figure 6 Apollo 9 Z-Accelerometer Bias

Creep is time dependent and can be classed as either elastic or recoverable, and either plastic or permanent. The affects of both at a specific time affects the operation of the accelerometer.

An example of the possible effects of creep is provided by the sterilization conditions imposed on the Viking accelerometer. Effects of the 260-degree F for 600 hours is shown in Figure 7 for the bias term of Bell Aerospace accelerometers. This may susggest that sterilization may be a device for accelerated testing; however, not enough is known at the present.

Creep is also significant in vibrating string accelerometers where the ribbons in tension are vibrated. If the ribbon tensions under acceleration is great enough to cause creep, the differences of frequency will change and a false acceleration will be indicated. Statistical techniques (Ref 4) can be applied to determine the creep over a period of time and under certain conditions.

For long life consideration, microcreep may exhibit itself as the mass change of a gyro. The mass change exhibits itself as a change in drift, somewhat in the same manner as exhibited by the initial stages of a bearing failure. One phenomenon may be falsely interpreted to be another.

Another example of a possible microcreep problem, although not definitely attributed, is the shift out-of-tolerance in the accuracy of the accelerometer in the Lunar Orbiter Program (Ref 3). The loss of the accelerometer accuracy was due to "extended storage." It had been stored so long in the same position that the critical axis bias adjustment had changed. The corrective procedure in this case was to institute a procedure to exercise the accelerometer and change storage positions periodically.

d. *Bellows* - Bellows are a source of failure because they are a moving element. User and manufacturer experience indicate the use of proven bellows with sufficient capacity in the original design will provide satisfactory life. Change in the techniques and materials should be avoided. Early in the engineering of the Titan Lateral Acceleration Sensing Accelerometers, a Titanium bellows was designed and tested that developed leaks. Retaining the stainless steel bellows has proven satisfactory over extended periods of time. No failures of the bellows have occurred in the program.

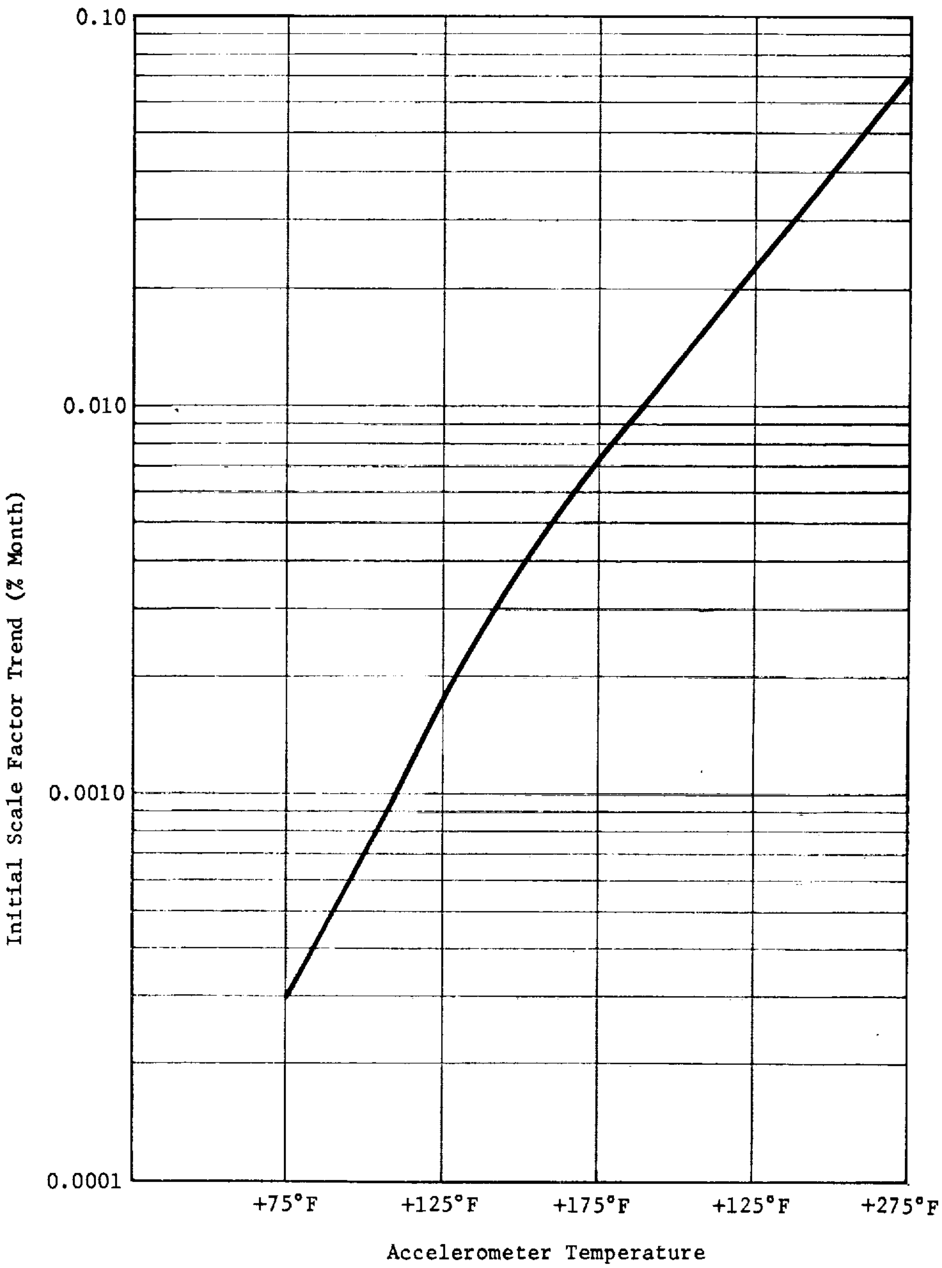

Figure 7 Typical Bell Aerospace Model VII Initial Scale Factor Trend (Aging) versus Accelerometer Temperature (Initial Refers to "New" Instrument)

2. Design Selection Criteria

a. Selection Criteria - The criteria are summarized in Table 5. The force feedback pendulous or suspended mass type of accelerometer is recommended for long life because of its simplicity. There is a wide range of manufacturing experience to draw upon and the performance levels are probably adequate for the future manned missions. Where higher performance is mandatory, the unbalanced gyro, or the PIGA with its life comparable to the gyro, must be chosen. The vibrating string accelerometer has not been applied in recent years and is considered to possess, at this time, inadequate experience for application to long life.

Table 5 Design Factors for the Long-Life Assurance of Accelerometers

Design Factor	Remarks
Simplicity*	Use the force feedback pendulous or suspended mass type for simplicity, all other parameters equal.
Higher Performance Mandatory	Employ unbalanced gyro or PIGA when the increase in complexity dictates the more complex device. The performance difference between the force feedback pendulous and the PIGA is probably less than a factor of 10.
Experience	Use only proven designs and processes. Minimize risk by selecting a fixed design with a long successful operational history, preferably, in a use similar to the new application.
Fluid	Fluid being a transport medium for dirt should be controlled both for particulates and effects of chemical reaction with other materials in the accelerometer subsequent to manufacture.

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*Solid state accelerometers are the simplest in principal and construction, but their lower accuracy has not permitted their use as navigation instruments. They should be used when performance requirements permit.

The design of an accelerometer is highly dependent upon the demands of the application in terms of the accuracy and the environment of the space vehicle. Table 6 presents some design configuration and application information for orientation purposes. In the force feedback accelerometer (proof mass or pendulous), the resolution and accuracy is related to the gain in the feedback loop, the resolution of the pickoff, and inversely proportional to the proof mass itself. The bandwidth is related to the square root of the gain in the feedback loop divided by the mass. Noise in the system is dependent on the bandwidth. All of these are design factors to be considered in the individual applications.

The following is extracted from Reference 5:

"No application has been reported where a successful accelerometer on one system has been used, without any change, on another system. Therefore, in applying an accelerometer to a system, some changes must be accepted. There is always some risk associated with these changes and this is usually identified as *Design Risk*.

"This *Design Risk* can be minimized by selecting a fixed design with a long successful operational history. The changes that are required to adopt the accelerometer to the new situation can be readily identified. Once these are established, the test program developed for the accelerometer should include an evaluation of these identified changes.

"Prior to actual use, data should be sufficient to allow a complete evaluation of the accelerometer which should include the effect of changes that had to be accepted to adopt it to the new application.

"Changes in approved practices or materials must be treated with extreme caution. History is full of situations where an undetected change in a process of material has cost millions of dollars to correct.

"Traceability requirements should be imposed early in any instrument program. This prevents acceptance of marginal units and can be used to trace the extent of a faulty assembly process or material used in instrument build."

Table 6 Manufacturers Number and Application

Model	Application Examples	Type	Temperature Range, °F	Proof Mass Sequence	Material of Proof Mass	Pickoff Type	Torque Type
Kistler 303B	Intelsat Tiros	Force Balance					
United Controls 5706	ATS						
United Controls	Titan III	Torque Balance		Pivot Jewell		Induction	Permanent Magnet
Bendix AB3-KB	Saturn	PIGA	04 ±6	Gas	Beryllium	Moving Coil	Permanent Magnet Servomotor
Delco PIGA	Titan	PIGA					
Delco Model AC613	Titan	Force Balance		Spring			Permanent Magnet
Sperry 16 PIPA	Apollo (CSM, LEM)	Pulsed Integrating Pendulum		Pivot & Magnetic	Beryllium	Magnetic	Permanent Magnet
Kearfott 2401	Mariner, LEM-Abort, Delta	Torque Balance	150 ±5	Hinged Pendulum		Inductive	Permanent Magnet
Systron Donner 5450	Biosatellite	Torque Balance		Pivot Jewell		Capacitive	Permanent Magnet
Systron Donner 4310	Surveyer	Torque Balance	10 -200	Pivot	Aluminum	Capacitive	Permanent Magnet
Systron Donner 4810	Titan III Apollo (CSM)	Torque Balance					
Statham A-304C	Saturn (Load)	Spring Mass					
Arma D5C	Atlas	Vibrating String	0 -70°C ±1°C	String	Beryllium	Magnetic	
Bell IXB	Viking	Torque Balance				Magnetic	
Bell IIIB/ VIIB	Ranger, Agena, Scout	Torque Balance					
Honeywell GG116	Gemini, Centaur	Torque Balance					
Honeywell GG177	Prime, Agena, Delta, Burner II, Centaur, Thor	Torque Balance	170°F Nominal				

The initial testing must be sufficiently complete to "weed out" the initial failures. The conventional "bath tub" failure curves apply to accelerometers. The manufacturing and test history must be sufficient to attain the minimum flat portion of the curve prior to the actual use in the space vehicle. The point of the minimum rate of failure is a quality that almost always is a result of experience with a particular design and specification. For example, the same accelerometer, insofar as the design is concerned, may be adjusted to different specifications.

In accelerometer design for long life, "the devil you know is better than the devil you don't know," is a basic guideline. Extremely exacting design, design analysis, process control, manufacturing control, test, and application is a mandatory requirement to realize the performance potential over the extended period of time. These principles, once applied to the force feedback proof mass or pendulous accelerometer will result in an accelerometer with no known life limitation over the time periods of concern in this chapter.

b. *Survey Results* - Two surveys of manufacturers and users were conducted. The first survey obtained opinions about the general life limiting problems and solutions of accelerometers. The second survey concerned specific accelerometers and how their reliability and lives were obtained. The comments and consensus of opinion of the first survey are given below. The results of the second survey are shown in Table 8.

1) *First Survey* - The accelerometers of the type being considered in this chapter are unique components. Each accelerometer is constructed with a design unique to the manufacturer with the exception of the MIT design PIGA and PIPA. Industry surveys are difficult to make on this type of component because the response tends to be somewhat biased to the particular manufacturers or users item. Nevertheless, there have been certain comments and consensus derived. The general consensus are:

1) There is no definitive element in accelerometers that is life limited. Materials used have lives beyond the 10 years life objective;

2) Periodic calibration is required to account for long term minor changes due to permanent magnet permanence and other unexplainable changes;

3) Very careful design and manufacture is important to long life achievement;

4) Trend analysis, that is, scale factor tests to determine the aging effects of the permanent magnetics are a requirement for precision pendulous accelerometers;

5) Once the initial design of the accelerometer satisfies the requirement for stability in the short term; the same steps required for that stability tends to build into the accelerometer long life capability.

The comments of those surveyed are summarized in the following paragraphs.

Brad Sage, Systron Donner - Telephone: (415) 682-6161

No problem in long life application of the Systron Donner genertic type 4310. (Note: the other type number in Table 6 are versions of this type; both single-axis and two-axis versions.) The stability of a one-year period is 0.05% and 0.005%. The accelerometer has accumulated 25,000 hours of vibration testing with no failures. The specimens were subjected to 2 g continuous vibration while running at 30 cps. The seal is soldered. No microcreep problems were observed. Life problems, if any, are externally induced by the electronics and other external influences and not the accelerometer instrument itself. The aging of the permanent magnetics are a factor, but a predictable factor.

The Systron Donner accelerometers (1-g range) have been installed in earth dams and run continuously for up to six years with no failures.

Aaron Copeland, Bell Aerospace

The Model IX is the latest of a series and is used in the Viking program. The output of the pickoff is amplified and fed back in a pulse loop. The accelerometer is the pendulous type and the pendulum is fluid immersed. Mr. Copeland knows of no life limitation problems. Customers have been operating the accelerometer continuously for five years. The flexure is elgiloy with high modulus of elasticity. The torquer is of ALNICO and requires periodic calibration to account for aging affect of the magnetics on the scale factor. The fluid used is silicon (510 and 55) and is inert. There are no components that wear.

James Davies, Kearfott Singer - Telephone: (201) 0256-4000,
Ext. 2257

Life of the Kearfott 2401 is essentially infinite. The Kearfott accelerometers are used both heated and unheated. Mr. Davies believed the shelf life to be well over 10 to 12 years. The experience on Subroc was good. The instruments are tested every three to six months over a period of many years. The fluid in the Kearfott accelerometers is the inert silicon. The seals are Viton, a silicon base rubber; and there is not a life limit on the seals. Mr. Davies offered the opinion that; "I would feel better about flying one five years old than a new one."

G. Fairweather, MIT - Telephone: (617) 864-6900, Ext 4029

The MIT design is the PIPA. This is essentially a gyro without a wheel and with a solid unbalanced float. The device has had no life problems. The bellows used are stainless steel, large, and present no problem. The fluid cleanliness process is important to avoid contaminates. The accelerometer is tested for float freedom. The magnetics are subject to a 30-day aging and are temperature compensated in the accelerometer. Most of the units operate at 130 degree F, but can operate at other temperatures. No problem with the Viton A rubber seals. Mr. Fairweather stated that the general performance capabilities of the PIPA is a level above the pendulum accelerometer.

Messrs. R. Maginn and Ivers, Martin Marietta Aerospace

Martin Marietta's Denver Division has been supplied accelerometers for application in their products by United Controls, Kearfott Singer, Systron Donner and Bell Aerospace Corporation (through United Aircraft). Martin Marietta user experience is extensive. There has been no identified problem with life except for the aging of the permanent magnets. Early in a design phase, titanium was unsuccessfully experimented with for the bellows in the Kearfott 2401; however, the stainless steel bellows proved satisfactory. The bellows is usually supplied to the accelerometer manufacturer by a separate company. In the Viking application there is a sterilization requirement of 600 hours at 275°F. There is some creep during this process, however, this has the effect of stabilizing the instrument. No problem was experienced with a 1200-g-pyro shock and a 10-g rms vibration on the Viking application.

In a review of the failure data on the lateral acceleration sensing system (LASS) used on the Titan III vehicle which incorporated the Kearfott Singer 2401 type accelerometer, only once was an accelerometer rejected for aging effects upon the scale factors (Reference 6). It is noteworthy that the life specification has been prolonged to seven years as actual test experience was accumulated. Data on the bias and scale factor on those systems is given in Table 7, and Figure 8 and 9.

2) *Second Survey* - The results of the second survey on specific accelerometers are summarized in Table 8.

c. *Alternate Approaches* - The general accelerometer technology on the force feedback pendulous type accelerometer has been developed to a state where long life is inherent in their design. The steps in the design required for the initial stability are those required for long term life performance. The operational simplicity of the force feedback accelerometer, a single moving element held in a nearly fixed position by a single feedback loop, makes it a highly desirable design.

When alternatives are considered, the alternative of a dry accelerometer should be made. The conventional accelerometer is wet, fluid filled with the wet fluid providing some of the damping for severe environment. The fluid provides an additional vehicle for particulate and chemical contamination. Therefore, there is a desirability for eliminating the wet fluid.

The dry accelerometer utilizes a concept where the damping is derived from a squeeze film concept. An oscillating disc that is a part of the displacement sensing system provides the required squeeze film gas damping. This concept requires another operating element, that is, the squeeze film oscillation system.

New concepts may be contrary to the depth of the experience with some of the existing designs. The principle drawback is the lack of experience data on the new concepts. The risks of applying a new concept with some possible advantages of long life must be traded against the designs with long experience in the desired operating conditions.

Again, existing technology appears to be sufficient for long life application unless a new performance requirement exists such that existing designs are not sufficient.

Table 7 LASS Performance History

Ser. No.	Axis	Bias ΔK_o (%)	Scale factor $\Delta\beta$ (ug)	Period (years)
1	A	0.051	213	1
	B	0.068	132	
6	A	0.028	157	1
	B	0.023	84	
7	A	0.018	193	1
	B	0.003	51	
8	A	0.011	236	1
	B	0.023	86	
9	A	0.054	35	1
	B	0.023	61	
13	A	0.023	178	1
	B	0.013	211	
16	A	0.008	120	1
	B	0.018	155	
17	A	0.026	111	1
	B	0.021	49	
18	A	0.012	69	1
	B	0.003	254	
21	A	0.074	107	1
	B	0.045	127	
28	A	0.020	101	1/2
	B	0.020	135	
29	A	0.005	97	1/2
	B	0.015	56	
8	B	0.054	100	4
11	A	0.002	1	3
	B	0.017	60.6	3
13	A	0.133	360	4
	B	0.141	34	4
19	A	0.056	249	4.5
	B	0.035	6.63	4.5
24	A	0.019	+681	7.25
	B	0.024	+113	7.25
27	A	0.055	-616	7.25
	B	0.023	+523	7.25
30	A	0.029	-25	1.0
	B	0.027	-6	1.0
31	A	0.013	+9	1.25
	B	0.008	-62	1.25
32	A	0.018	-84	2.0
	B	0.022	+186	2.0
33	A	0.041	-320	2.0
	B	0.002	+175	2.0
34	A	0.002	+3784	2.0 NOTE:
	B	0.018	-3580	2.0 Scrapped
35	A	0.018	-70	1.0
	B	0.009	-22	1.0
36	A	0.017	-250	1.0
	B	0.025	+27	1.0

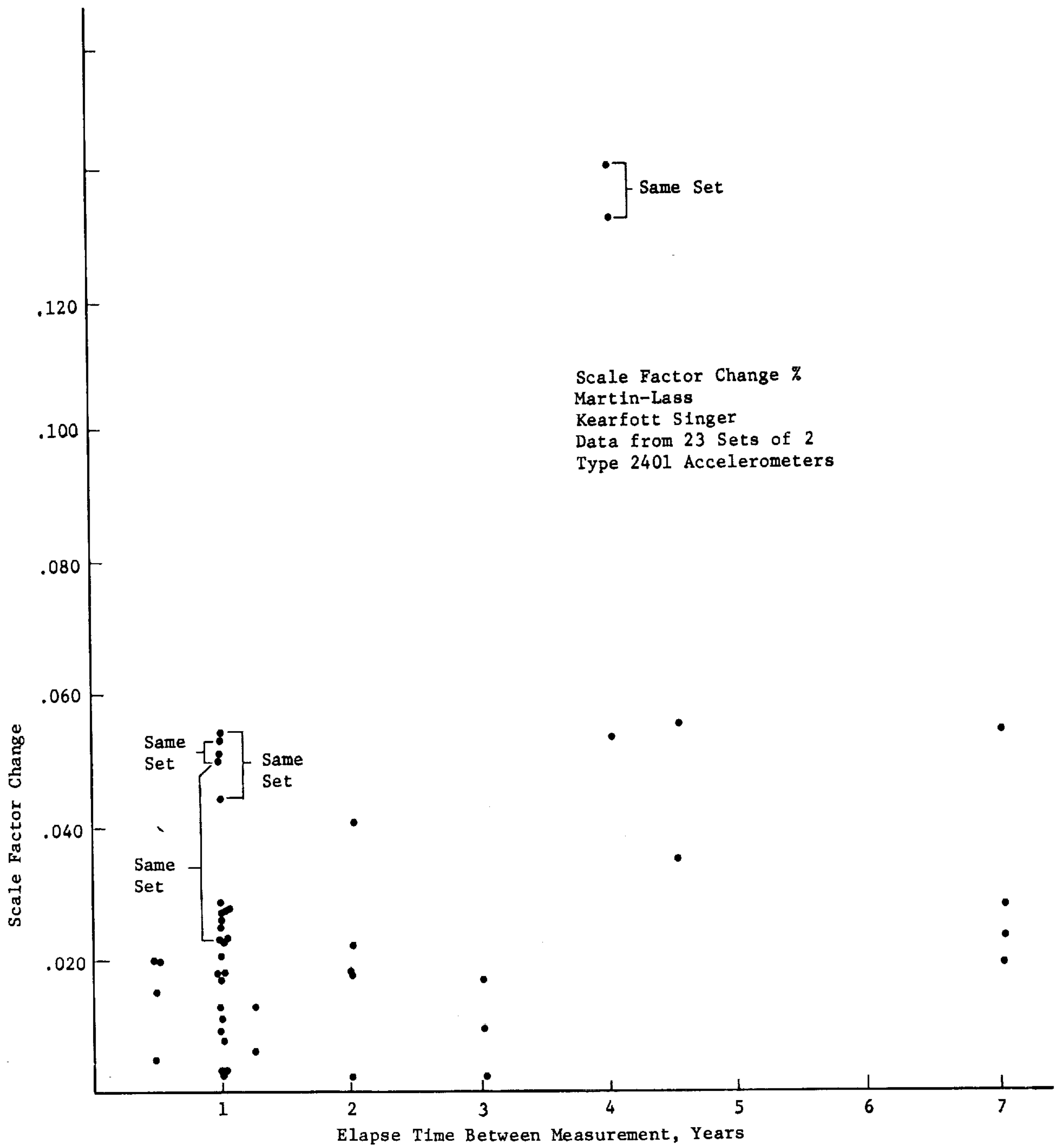


Figure 8 Scale Factor Change vs. Elapsed Time

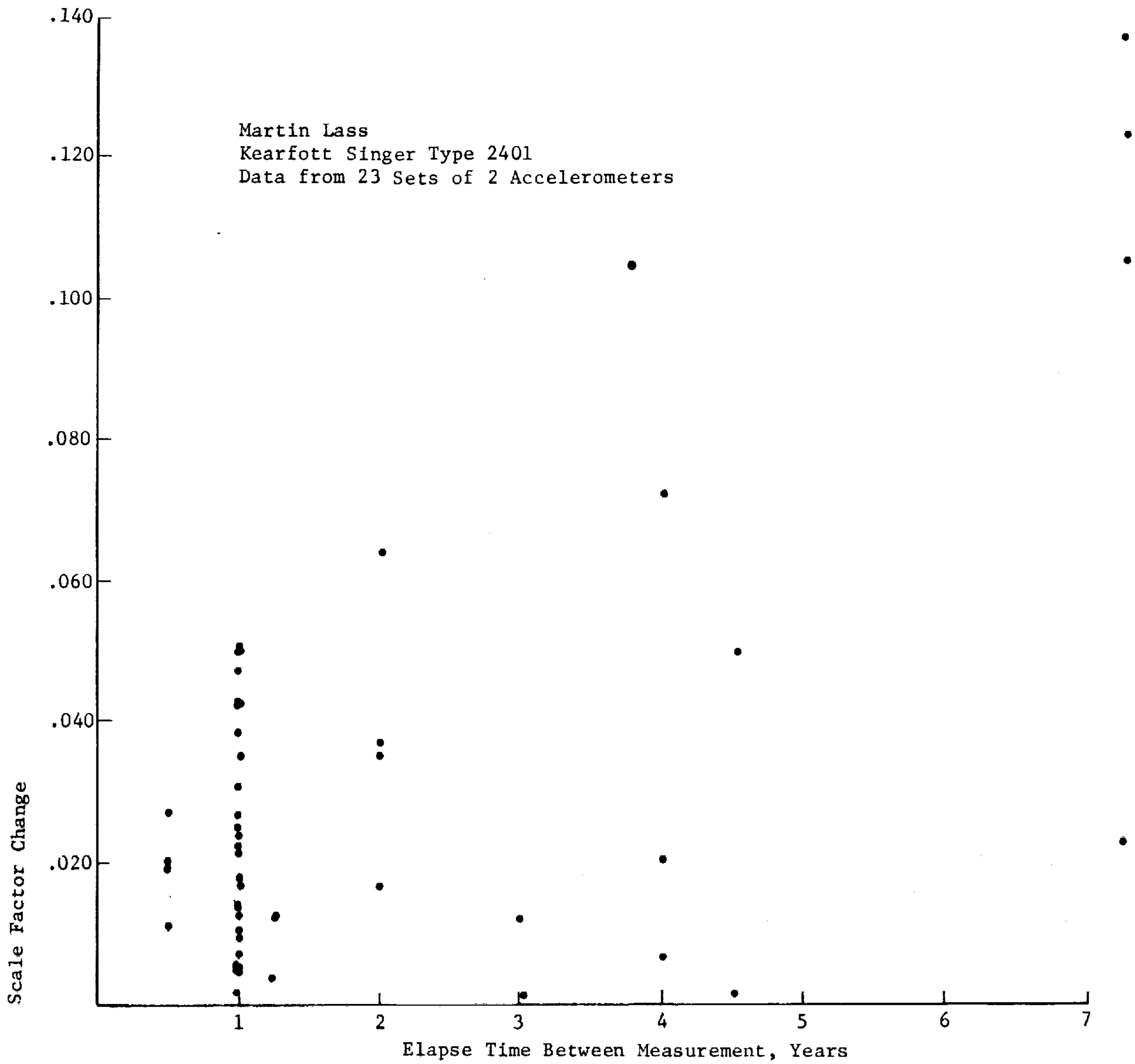


Figure 9 Bias Factor Change vs. Elapsed Time

Table 8 Survey of Users and Manufacturers on Specific Parts

Part Description and Using Program	Users Opinion of Why Part Was successful	Manufacturer's Opinion of Why Part Was Successful	Recommended Guidelines Improvements, Procurement Spec Requirements, etc	Relative Impact on Yield & Cost of Incorporating Recommendations
Lateral Acceleration Sensing System PD96S0007 (2 Accelerometers)	User believes good specification, good coordination with vendor, and good margin of performance in the sensor made the device a successful item.	Manufacturer believes: 1. Because the accelerometer performance capability far exceeded the requirement, the likelihood of a performance failure was practically eliminated. 2. Lot traceability and screening of parts payed off. 3. In most programs competitive bidding would have eliminated the performance margin advantage. 4. Beyond documentation, the ability of the user and manufacturer to solve development problems finds its way in the final product integrity.	1. Good margin of performance. 2. Lot traceability and screening.	1. Moderate cost increase. 2. Some cost increase.
Kearfott 2401 Mariner-Titan	Unit performance specifications exceed system requirements.	Accelerometer specifications far exceed system requirements; good basic design.	Use good proven design with large performance margin.	Moderate increase in cost.
Systron-Denner 4810 Surveyor	Some development (Q.C.) problems, less design margin, no problems after Q.C. bugs worked.	Good design, good quality assurance, lot control.	Incorporated life and reliability requirements in the design specifications.	Possibly increased cost.
Honeywell GG 177 Agena	Long development program - many produced.	Stringent development program, process controls.	Require acceptance screening at sufficient stress levels to assure quality.	Lower yield, higher cost.
United Controls 5706 Titan IIIB	Low system performance requirements.	Simple basic design, good performance margin over system specifications.	Review reliability assurance and reporting systems to assure adequacy.	Increased cost reflected to unit.
Delco 653A Carousel IV	Many built, no development required.	Manufactured in large quantities, established production techniques, good Q.C.	Use procedures which minimize degradation and possibility of damage during test.	Increased cost.
Kearfott 2414 A-7 SRAM	High production quantities; good basic design.	Good basic design, long development program, many units produced.	Nothing wrong with design have safety margin between unit spec and system requirements.	Increased cost over "low cost" unit.
Bendix PIGA Pershing	Extensive development program.	Stringent controls exercised during all phases of build.	Require adequate controls over materials, procedures, adequate screening.	Lower yield, increased cost.

d. Hardware Life - Operating or non-operating the force feedback pendulous accelerometer is good for 10 years. The only known life limiting and aging elements are the permanent magnets. Periodic adjustment for scale factor can be made because there is a predictable log decay of the scale factor. Periodic tests to detect scale factor change can also be used. If greater accuracy is required (factor less than 10) the PIGA can be used. However, the life will be limited to 3,000 to 10,000 hours for the ball bearing PIGA and 10,000 hours for the gas bearing PIGA. Vibrating string accelerometer is not recommended at this time because of a lack of recent experience data.

e. Application Guidelines - Because of the critical nature of performance, accelerometers will probably require periodic testing even though they inherently possess long life. The change in scale factor due to permanent magnet changes will, with time, require some adjustment.

It is important to choose an accelerometer whose performance margin or level exceeds the demands of the application. This may be more expensive at the outset; however, the priority should be applied to performance margin along with costs. With a performance margin, the instabilities that are always present to some degree can be below the specification limit.

D. TEST METHODOLOGY AND REQUIREMENTS

Depending upon the tolerance limits required in the particular application, a testing and recalibration process will probably be required to check and possibly calibrate the accelerometer. Testing at intervals may be required to determine the trend in the bias instability and the scale factor. Scale factor, a function of the aging of permanent magnets over the time period, can be predicted from initial data and subsequent test data prior to flight. However, for usage exceeding five years, periodic tests should be made. The general stability of the accelerometer can also be measured. Normally, only 40% of the cost of the accelerometer is in the initial manufacture. The rest of the cost is for testing and documentation. For the long term and repeated manned application, the testing effort will no doubt increase.

There is no reliable initial test process to enhance long life performance. There is no known accelerated testing technique on the complete assembly. The experiment with sterilization would appear to stabilize the accelerometer as seen by the data presented earlier in the report; however, there is no long term data on these accelerometers. This test process would be deemed too risky at this time.

Periodic testing involving repetitive testing may be required for scale factor and bias throughout the life of the accelerometer. The overall complexity of the test program is highly dependent on the accelerometer sensitivity and dynamic range. Normally, all the tests can be made at 1-g, that is, without the use of a centrifuge.

There is no screening processes or wear-in tests employed for accelerometers.

E. PROCESS CONTROL REQUIREMENTS

Accelerometer manufacture process control is:

1) Design dependent, and;

2) Usually is a result of evolution and experience.

Because of the proprietary aspect of the accelerometer, no general process control steps can be stated applicable to each and every accelerometer design.

Process control for contamination control, clean assembly areas, pre-conditioning of permanent magnets, and temperature cycling for material creep are all candidates for control during the manufacturing process.

F. PART USAGE

1. Previous Manned and Unmanned Accelerometer Applications in the Space Program

A review of the present space programs has yielded data on the application of accelerometers to the present manned and unmanned programs. The purpose is to provide a background of previous usage and information on the life aspects of previous designs.

Table 9 presents a list of the programs examined. Programs where accelerometers have been applied are underscored. Most of the satellites have been of the spinning type and did not utilize attitude control and inertial instruments. There are, however, significant applications and these are set forth in Table 10.

Considering the application of present accelerometers, only those that are directly applicable to manned flight will be included. These are usually associated with the following functions:

1) Inertial Navigation and Guidance - An inertial measurement unit, either gimballed or strapdown, is used to provide acceleration signals. These are normally processed in computer logic to obtain velocity and/or position for navigation and guidance. Boost acceleration levels are of the order of 8 to 10 g on the smaller vehicles. The accelerometer accuracy requirements are on the order of 10^{-5} g for bias uncertainty, and 10^{-5} g/g for scale factor uncertainty;

2) Velocity Change, V (course correction and space navigation) - Velocity Changes are used for a guidance maneuver to determine the appropriate engine cutoff signal. The acceleration level during the maneuver is generally on the order of tenths of a "g" and the overall accuracy demanded on the acceleromter measurement is on the order of 10^{-4} g;

3) Control - Load alleviation in large launch vehicles and damping of nutational motion in spinning vehicles involve levels generally less than a "g" and accuracies on the order of 10^{-2} g.

Table 9 Space Programs Reviewed

An underline indicates that program employed accelerometers.

Score	Echo(s)	Oscar
Courier	Telstar(s)	Relay(s)
Symcom(s)	Loft(s)	LES(s)
Early Bird	IDCS(s)	<u>Intelsat(s)</u>
<u>ATS(s)</u>	TACSATCOM	Sye Net
DCSS(s)	<u>TIROS(s)</u>	Nimbus(s)
ESSA(s)	TRANSIT(s)	NAVSAT
SECOR(s)	BEAGON	EXPLORER(s)
OSO(s)	AOSO(s)	OGO
<u>Mariner</u>	VELA	Pioneer
OAO	Lunar Orbiter	<u>Biosatellite</u>
<u>Surveyer</u>	<u>Ranger</u>	<u>Gemini</u>
<u>Scout</u>	<u>Delta</u>	<u>Apollo</u>
<u>Titan</u>	<u>Centaur</u>	<u>Prime</u>
<u>Saturn</u>	<u>Atlas</u>	<u>Agena</u>
	<u>Burner</u>	

Table 10 Accelerometer Applications

VEHICLE	ACCELEROMETER	APPLICATION
Intelsat	Kisler 303B	Nutation Damping
ATS	United Controls 5706	Nutation Variable
Tiros	Kisler 303B	
Mariner	Kearfott 2401	V
Biosatellite	Systron Donnor 5450	Boost Thrust
Surveyer	Systron Donnor 4310	V
Ranger	Bell IIIB	V
Gemini	Honeywell GG116	IMU
Apollo	Sperry 16 PIP	IMU
LEM-Abort	Kearfott 2401	Strapped Down
Prime	Honeywell GG177	Strapped Down
Agena	Honeywell GG177	V & Strapped Down
Burner II	Honeywell GG177	V
Centaur	Honeywell GG116	IMU
Centaur	Honeywell GG177	IMU
Titan	Kearfott 2401	Control
Titan	Systron Donnor 4810	V
Titan	United Controls	V
Saturn	Bendix AB3-KB	IMU
Delta	Honeywell GG177	V
Titan IIIC	Delco PIGA	IMU
Titan IIIC	Delco AC653	IMU
Delta	Kearfott 2401	IMU (Strapped Down)
Thor	Honeywell GG177	V
Saturn	Statham A-304C	Control
Atlas	Arma D4C	IMU
Viking	Bell IXB	Control

Accelerometers are commonly employed in several other functions in addition to the above. These include leveling of an inertial platform prior to launch and monitoring for flight instrumentation. Also, in some of the more demanding applications, accelerometers are used in instrumenting the performance of electric propulsion for spacecraft. This later application is rather interesting since the requirements involve active measurement for periods of months. Consequently, it requires, in addition to the extremely low levels of acceleration measurement, very good stability. Electric propulsion at this time does not involve manned space vehicles.

Table 6 sets forth the manufacturer's model number, name of the program that utilizes the instrument and presents some of the pertinent characteristics of the accelerometer (Ref 5). The specific operating parameters of each accelerometer instrument that is not directly involved with life is not provided. It should be stated again that the technology is in a state of flux. Some concepts, like the vibrating string accelerometer, apparently are not applied anymore and constitutes obsolete designs while others are emerging and may be ready for application.

G. GENERAL DESCRIPTION OF ACCELEROMETERS BEING CONSIDERED

Accelerometers are measuring devices that provide an output corresponding to non-gravitational acceleration. They employ Newton's second law. The accelerometer instrument measures force per unit mass. When applied to a space vehicle, it measures the component along the sensitive axis of the instrument of non-gravitational forces. The flight computer derived gravitational acceleration is summed with the instrument measured acceleration to obtain inertial acceleration for use in navigation. See Figure 11. Velocity and position is derived by the integration(s) of the initial acceleration.

There are several fundamental different types of accelerometers commonly employed. These are:

1) Pendulous;

2) Vibrating String;

3) Pendulous Integrating Gyro Accelerometer (PIGA).

1. The Pendulous Accelerometer

Figure 12 sets forth a basic design of the pendulous type of accelerometer. The most common accelerometer at the present time is the pendulous type. It is used in a force rebalance electronic mode where the output position of a "proof mass" away from the null initial position is amplified and fed back to a torquer. The current to the torquer in this mode is the measure of acceleration (force/mass) of the proof mass pendulous element. The entire flexure supported elements are usually immersed in a fluid in applications involving vibration and shock. This fluid also serves as a damping medium.

The basic elements of the pendulous accelerometers are:

1) Pendulous Proof Mass;

2) Flexure Suspension;

3) Torquer (usually permanent magnet);

4) Pickoff-aircore differential transformer or capacitive;

5) Fluid Damping (optional).

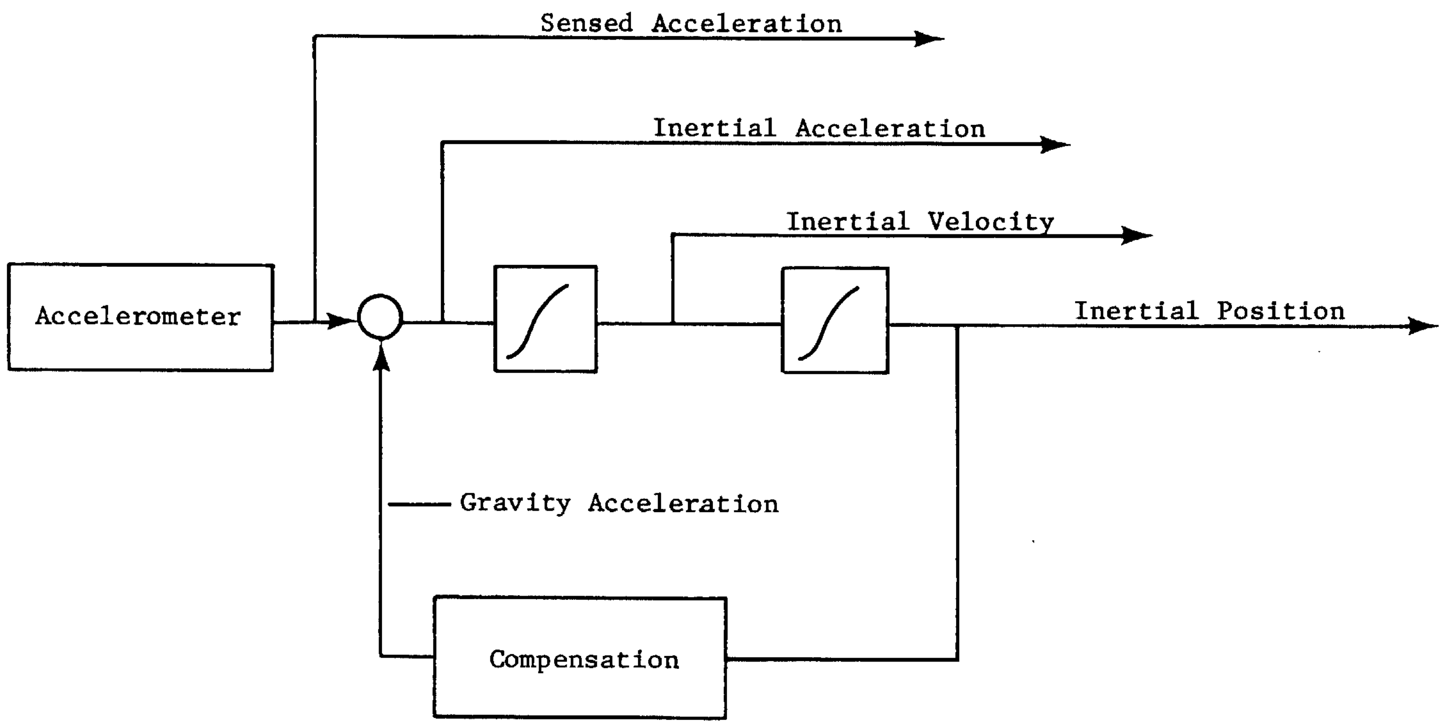


Figure 11 Typical Navigation Loop

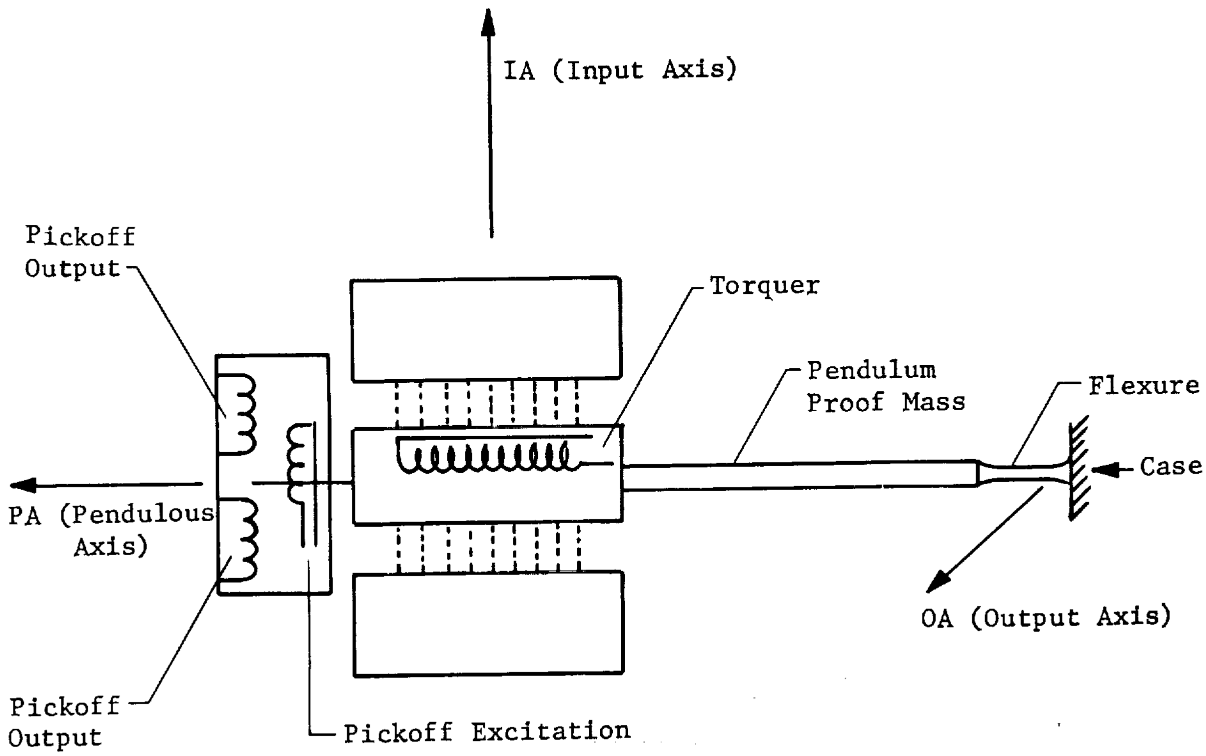


Figure 12 Pendulous Accelerometer

2. Vibrating String Accelerometer

This accelerometer involves a seismic mass supported by two flexible tapes that vibrate with frequencies proportional to the tensions applied. Under acceleration, the seismic mass causes an increase of tension in one tape and a decrease in the other. The frequency difference between the two tapes is proportional to the applied acceleration. See Figure 13.

3. Pendulous Integrating Gyro Accelerometer (PIGA)

This type is considered to be the most accurate device in operational use. A floating single-degree-of-freedom gyro designed with pendulousity along its spin reference axis serves as the force sensor. Acceleration along the gyro input axis acts on the resulting unbalance to develop a torque that rotates the float about its output axis. A turntable supports the gyro case and is driven by a torquemotor to rotate the gyro about its input axis. When driven at the proper rate by a servoloop, the gyroscopic precession torque is equal in magnitude and opposite in direction to the pendulosity torque. Output is obtained from an angular encoder on the turntable shaft which produces a pulse rate that is proportional to the acceleration.

The PIGA takes advantage of a direct implementation of Newton's second law. It possesses the characteristic of linearity and low hysterises. The gyro itself is comparable in complexity to other precision gyros with the addition of a servomotor and the digitizer pickoff (slip rings and bearings).

The unbalance gyro accelerometer is heavier and more costly when compared to the more simple pendulous type. See Figure 14. The unbalanced gyro accelerometer as an inertial instrument is sensitive to input angular rates relative to inertial space. A consideration that becomes of increased importance in a strapped down application.

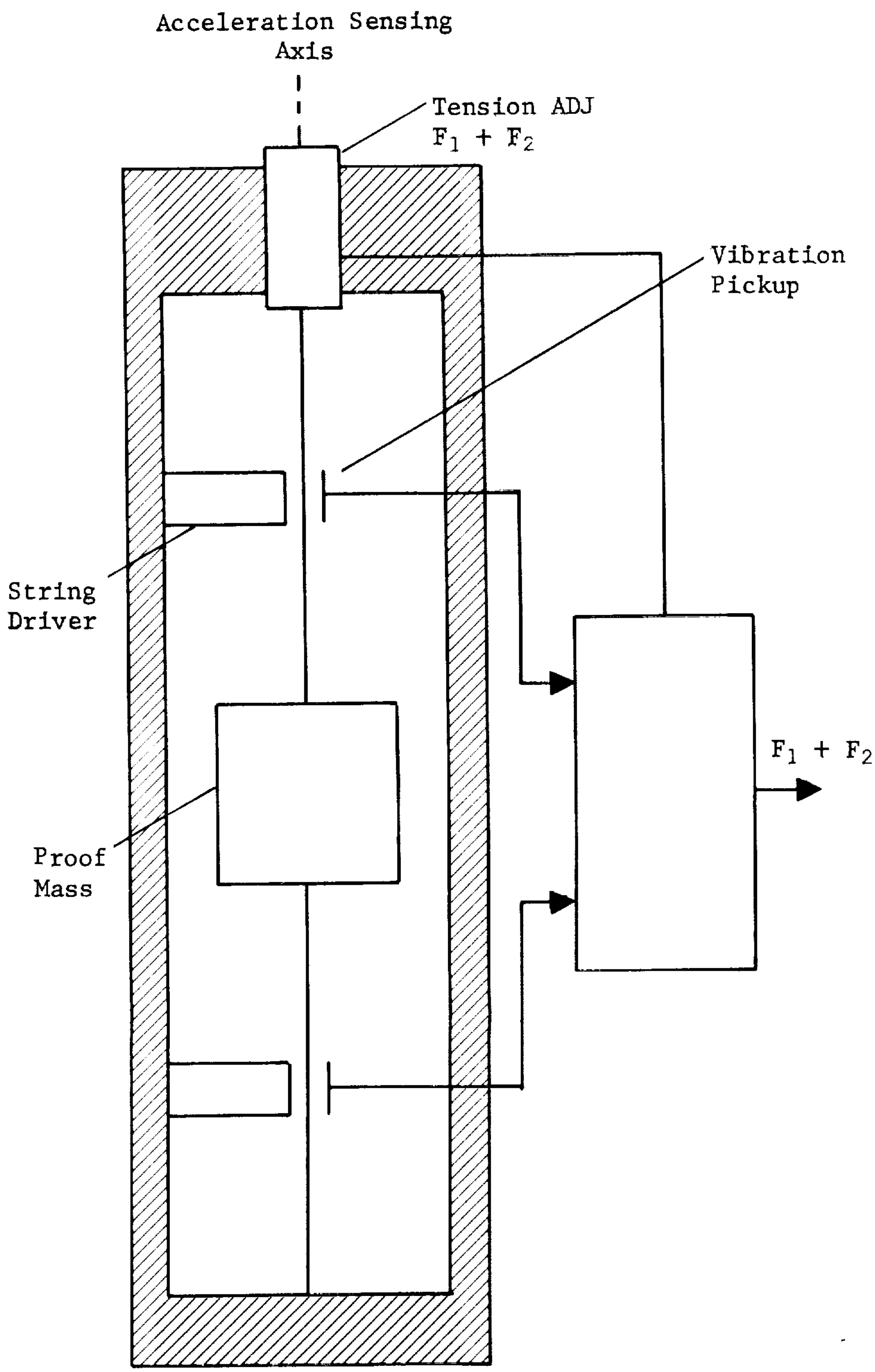


Figure 13 Vibrating String Accelerometer

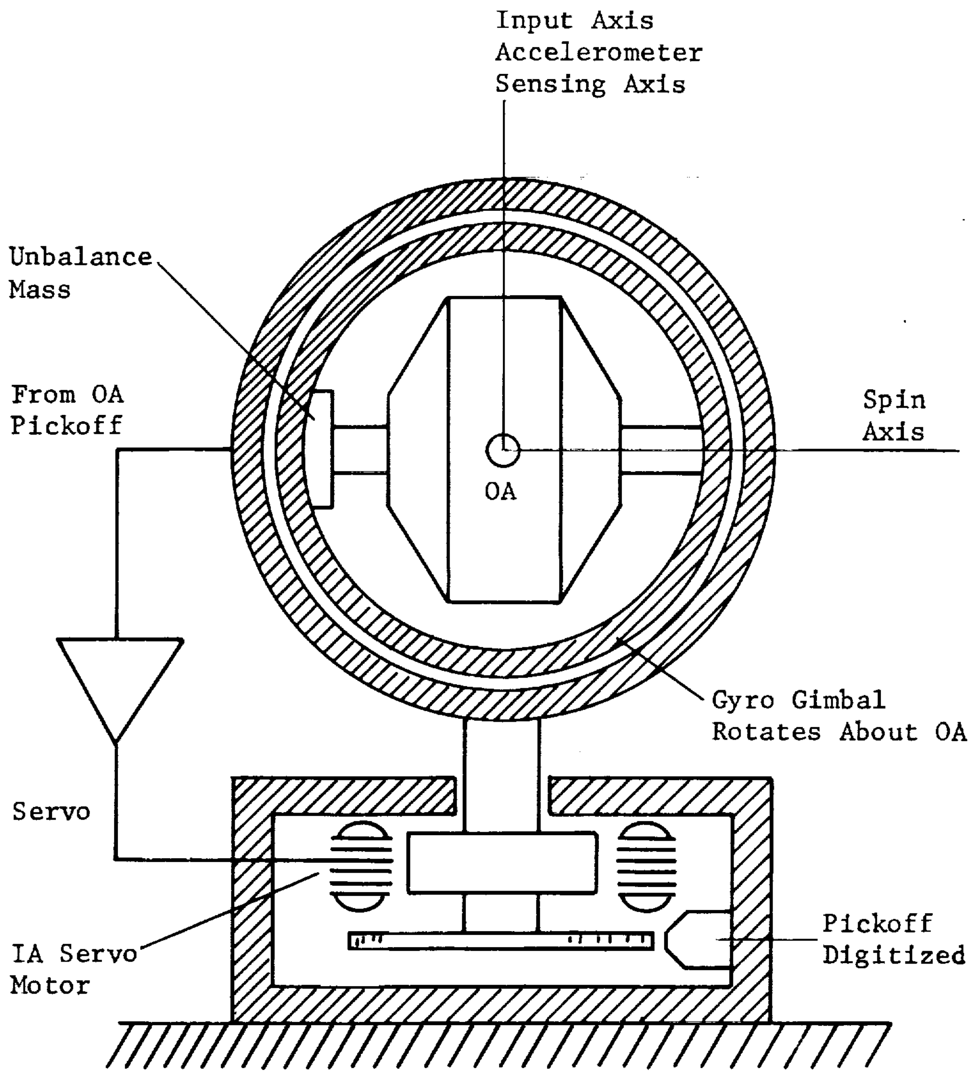


Figure 14 Unbalanced Gyro Accelerometer

Another type of accelerometer is the PIPA, for Pulsed Integrating Pendulous Accelerometer. The design of this instrument is most like the unbalanced gyro without the wheel and the servoplatform loop. In this instrument there is a pulse rebalance feedback loop between the output axis pickoff and a torque motor on the output axis and a combination of fluid and magnetic flotation. Very careful handling (for example, constant heat) may be required for high accuracy and stability for the PIPA instruments. References 5 thru 11 provide additional descriptive information.

There are many other types of instruments for sensing acceleration. It could be concluded that the state-of-the-art is in a state of flux and there is no single configuration applied.

H. REFERENCES

1. Rollin J. Parker and Robert J. Sludder: *Permanent Magnets and Their Applications*. John Wily and Sons, Inc.

2. *The Flight of Apollo IX and the Debut of Strapdown Guidance in Space on Manned Flight, Vol VI*. Hamilton Standard, Farmington, Connecticut, SP02069.

3. Butterfield and Herz: *Experience Applicable to the Viking Lander from a Study of Related Space Flight Projects*. General Electric Company, NASAC12-111876.

4. *Accelerated Life Testing of Guidance Components*. Contract AF33/615-1157, Autonetics Division of North American Aviation, Inc.

5. *Space Vehicle Accelerometer Applications*. Draft of Monograph, March 1972, NASA.

6. *Radio/Optical/Strapdown Inertial Guidance Study for Advanced Kick Stage Applications, Vol II*. TRW Systems Group, Redondo Beach, California, 30 June 1967.

7. Miscellaneous Failure Reports, Martin Marietta Aerospace, May 1972.

8. *Inertial Navigation Systems and Components*. AGARD, May 1968.

9. G. Bukou: *The PIPA*. MIT Instrumentation Laboratory, September 1968.

10. R. L. Gates: *Reviewing of the Status of Inertial Sensors*. Martin Marietta Aerospace, Control Engineering, March 1971.

11. G. R. Pitman: *Inertial Guidance*. John Wiley, 1962.

IV. GYROSCOPES AND BEARINGS

by R. Opper

IV. GYROSCOPES AND BEARINGS

A. INTRODUCTION

The gyroscopes considered are the sensing displacement and rate gyroscopes and the control moment gyroscopes. Gyroscopes are a vendor unique item; that is, each manufacturer's design is somewhat different from another manufacturer's design. There are some elements in nearly every design that are not common. Because the exact unique gyroscope design features that future manned vehicles may use are unknown, this report highlights those design features where there is a degree of commonality. It also highlights those features that are involved in enhancing the performance potential for the extended period of time. Section G of this chapter presents a general description of gyroscopes for reader information.

Experience indicates that the primary life limiting part of gyros is the motor/rotor bearings. This is followed by drift instability, contamination, leaks and pivots. This report recommends further investigation into a particular design of the grease bearing concept as a solution to some of the life problems of the conventional ball bearing and the gas bearing gyroscopes; the grease bearing has a greater tolerance to contamination as compared to the gas and ball bearing. The grease bearing also possesses a potential for extremely long life. Generally, the ball bearing gyro wheel can be expected to achieve an operating life of 3000 to 6000 hours under service conditions; the gas bearing life is around 10,000 hours and the grease bearing is potentially much greater.

Long-life is influenced by many factors. Exacting design, manufacturing care and application are extremely important factors. For example: a piece of dust weighing a thousandth of a gram, or a creep in the material of a gyro float, can introduce a drift rate equivalent to a nautical mile in an inertial grade gyro. How the failure occurs is important as well as how the gyro can be tested to determine incipient failures and how the gyro can be screened to determine a good gyro from the life standpoint. Figure 1 illustrates how gyro performance and bearing life improve with maturity of design.

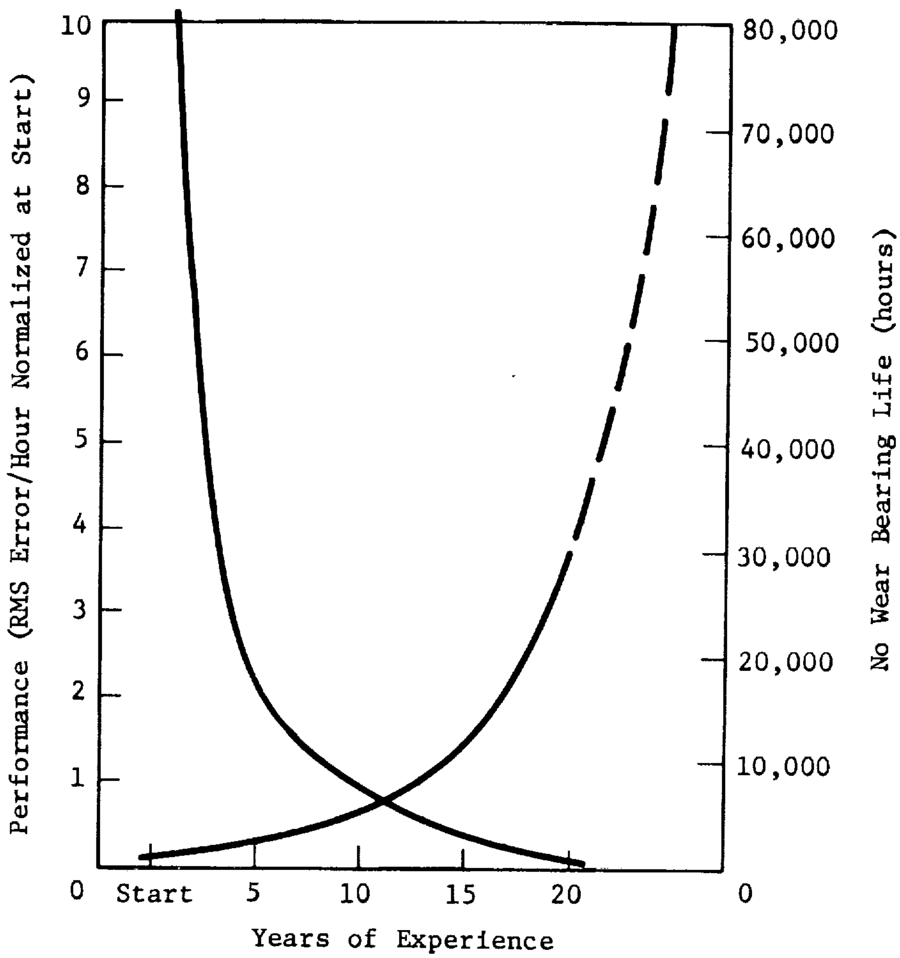


Figure 1 History of Gyro Performance and Life (Reference 1)

B. GUIDELINES FOR LONG-LIFE ASSURANCE

The major problem with gyroscopes is the wheel bearing reliability since its life ranges from 3000 to 10,000 hours. Testing of the gyros for incipient gyro failure during this interval of time is necessary in some applications. Other life limiting factors are the permanent magnet torquers, contamination, and instability in the material causing excessive drift rate.

1. Design Guidelines

1) If the mission requirement is for long-life with interrupted operations, especially in severe mechanical environments, the bearing choice favors a ball bearing wheel gyro because bearing failure takes place over a period of hundreds of hours and the ability to restart is predictable from test data .

2) If the mission requirement for long-life is for uninterrupted operation, the bearing choice favors the gas bearing because there is no wear under run conditions. If the application permits continuous operation, a gas bearing gyro should be used because it permits increased performance and minimal frictional wear in the bearing due to start-stops.

3) If there is a mature existing design that meets the identical new gyro requirements, use it in preference to a new design. A new development is a potential source for major problems.

4) Exacting design, manufacturing, and design for manufacturing is important for a successful product because material creep, chemical compatibility and design to prevent voids and sharp edges has proven necessary by experience.

5) Use the lowest operating temperature for the gyroscope commensurate with the application because chemical action in the lubricant decreases with decreased temperature.

6) Incorporate spin motor rotation detection in the design to permit run-down testing at the systems level.

2. Process Control Guidelines

1) Extreme cleanliness in the manufacturing process is essential because dirt particles as small as 0.001 grams can cause excessive drift in certain applications.

3. Test Guidelines

1) Monitor gyro parameter operating levels and stabilities to establish a record against which observed changes can be evaluated.

4. Application Guidelines

1) Be certain that the gyroscope specifications meet or exceeds the application requirements for all parameters. The most critical parameter pertaining to reliability are vibration, temperature, shock and acceleration.

2) Redundancy options must consider the more gradual failure of the ball bearing gyro as opposed to the almost instantaneous seizure when the gas bearing fails.

5. Special Consideration

1) The production run of wheels in any gyro program should include a quality monitoring plan in which sample wheels are operated to failure according to appropriately specified failure criteria. These wheels should then be inspected for the condition of the materials in the wheel assembly. The results which represent typical conditions of the entire run, provide a basis for extrapolation of wheel failure rates, operating life, and reliability factors. They also provide a basis for failure analysis, product and yield improvement, and comparison with similar records from other programs.

C. LIFE LIMITING PROBLEMS AND SOLUTIONS

1. Failure Mechanism Analysis

The life of a gyro is a function of how the gyro is used. The major life limiting part in the gyro is the motor/rotor bearing. The primary failure mode for a gas bearing is high friction. The friction in the bearing becomes too large for the starting torque of the spin motor to overcome, and the wheel fails to start. Consequently, there is a tendency to keep the gas bearing wheel spinning continuously. In ball bearing gyros, the most common failure mode is the wheel bearing becoming noisy due to lubrication breakdown. Table 1 present failure modes, effects, and detection methods.

The following paragraphs discuss bearing design, life, failure mechanisms and lubrication. Many of the life problems are substantiated by facts in the Previous Experience presented in Section H of this chapter.

Table 1 Failure Mechanism Analysis

Part and Function	Failure Mode	Effects	Probable Failure Mechanisms	Detection Method
Gyro Wheel Bearing (both ball bearing and gas)	Lubricant or running surface deterioration	Output noise to no output	Contamination, wearout	Noise on output
Fluid Contamination	Foreign particle in damping gap, gimbal pivots	Noise on output	Contamination	Float freedom drift stability
Fluid Leaks	Mass Unbalance	Excessive drift	Inadequate seal solder voids	Drift tests
Broken Pivots and contaminated pivots	High drift	Execssive drift	Contamination	Drift tests
Drift Instability	Material creep	Excessive drift	Material instability with time	Drift tests

There are primarily two methods of supporting the rotating wheel and providing the required mass stability of the wheel assembly:

1) Ball-bearing (elastohydrodynamic) Support, and;

2) Gas-bearing (hydrodynamic) Support.

Another method involving grease support (reference 1) is under study and development; however, there is no long-life experience available for this concept. This report recommends further work on the grease bearings.

In the ball-bearing support, the required stiffness is obtained by varying a combination of ball-bearing assembly design elements. In the gas bearing support, the required stiffness is obtained by varying the bearing gap geometry or gas density. Both types of support are complex with the limiting considerations being the available motor torque, required stiffness, thermal gradients, bearing operating temperature tolerance, lubricants, power requirements and heat-removal mechanisms.

Wheel bearing life is dependent on the stability characteristics of the bearing and its lubricant and is affected by the failure definition chosen. Life is properly defined as the operating hours during which the mass stability caused by the wheel remains within the system specification. However, ultimate functional failure occurs when the gyroscope for any reason will not satisfactorily perform its system function.

a. Ball Bearings - Experience on several programs large enough to produce reliable statistics shows clearly that a small percentage of selected ball bearings produced wheels that had a very long life (i.e., 30,000 hours or more). A larger percentage produced wheels which had a moderate life (i.e., 3000 to 6000 hours). A minimal percentage (more than 50%) did not produce good wheels. The population of gyros on these large programs contained a proportion of long-life and moderate-life bearings. The vast difference in life expectancy between the two groups explains why examination of the running characteristics of a gyro population during the first few hundred hours as practiced on these programs did not necessarily correlate with the eventual failure rate (Reference 2). Some gyro programs, (MIT/ILOAO) suggest that very long life and high reliability can be achieved through the use of adequate selecting techniques. Many techniques are available for screening bearings, retainers, and completed wheels to eliminate assemblies with short life. This screening can be done with high efficiency.

Bearing life may be improved by changes in operating temperature. Results from the Lunar Orbiter program, in which both Sperry SYG1000 and Kearfott Alpha gyros were employed, suggest that a 20°F reduction in operating temperature (to 145°F) resulted in doubled operating life (Reference 2). This can be attributed to the principle that the speed of chemical reactions decreases with decreasing temperature. Assuming no change in the lubricant, the improvement might also be atrributed to the increased elastohydrodynamic film thickness resulting from the increased oil viscosity at the lower temperature.

The spin-axis gyroscope bearing is a major determinant of both gyro life and performance. Fractional microinch position stability over extended time periods is required for the high performance space vehicles. To achieve this stability and to achieve long life, the ball bearing must run successfully on an elasto-hydrodynamic fluid film.

To achieve the current state-of-the-art bearing metallurgy, geometry, groove-surface topography and chemistry, lubrication ball-retainer, contamination control, dynamic behavior, testing and process variable have all been improved (Reference 3). Also involved are the surface-film-piercing asperity reduction, surface-chemistry improvement, and lubrication-mechanism advancement.

Of major concern in long-life assurance is the adaptation of measuring devices to join the functional tests in evaluation of: (1) bearing characteristics, (2) potential life and performance at various stages during the manufacturing process, and (3) during the periodic testing during the service life of the gyroscopes instrument. This failure relates directly to the "bath tub" failure relationship and the realization of the flat portion for long life application.

The principal requirements of the gyroscopic spin-axis ball bearing are:

1) Long life at the required performance level;

2) Freedom from physical and chemical degradation of all elements of the bearing, and;

3) The maintenance of dimensional stability of the gyroscope element.

Failure can be evidenced by a slight deterioration in the gyro performance to the outright inability of the spin momentum wheel to turn because of seizure. The dimensional stability of the bearing can be illustrated by the fact that, in a typical high performance gyroscope, the error specification can be 0.015°/hr for 0.2 microinch displacement in the wheel along the spin-axis (assuming a 2×10^6 gm cm/sec^2 wheel weighing 225 gm in a 1 g field).

1) Mechanism of Failure for Ball Bearings - The bearing must have a full stable elastohydrodynamic (EHD) film of lubrication for stable and satisfactory performance. This is fundamental. Piercing of this film during running causes chemical and physical degradation of the lubricant and of the metal surface, this in turn changes the location of the wheel. As stated previously, deviations in the location of fractions of microinches can cause the gyroscope to possess excessive errors. The piercing of the lubrication film is a progressive action once started. As the debris formed by the high speed metal contact progresses, further piercing of the film occurs. The lubricant sludge formed by this process is deposited beside the pressure zones and has a sponge effect in drawing the lubricant from the region where it is needed to maintain a film. As the mode of failure progresses, bearing torque becomes erratic, the lubricant is dissipated, the metal wears, and the torques increase to the point and condition where the speed is decreased. The running time from the onset of the piercing of the lubricant film to the condition of speed change can be several thousand hours. Metal fatigue does not play a major part in gyroscope bearing failure.

Proper initial design enhances the operating of the bearing and the maintenance of the EHD film. These design factors are:

1) The metal components (races and balls) must have a geometric form that generates the required EHD film with acceptable stress levels over the entire pressure zone.

2) The metal must sustain the load without plastic flow or surface damage.

3) The surfaces must be free of film-piercing asperities and must chemically support a boundry lubricating film at low speed and an EHD film at operating speed.

4) The lubricant must demonstrate the chemical and physical properties needed to achieve these films with acceptable torque level and possess both chemical and thermal stability.

5) The ball retainer must maintain a controlled lubricant reservoir and circulate this lubricant as needed for a full EHD film.

6) The environment of the bearing must be chemically, physically and thermally compatible with the bearing.

7) The bearing must be free of contamination that can cause piercing of the EHD film.

8) Insensitivity of the wheel location to acceleration field variations.

9) Stability of the bulk lubricant and film uniformity.

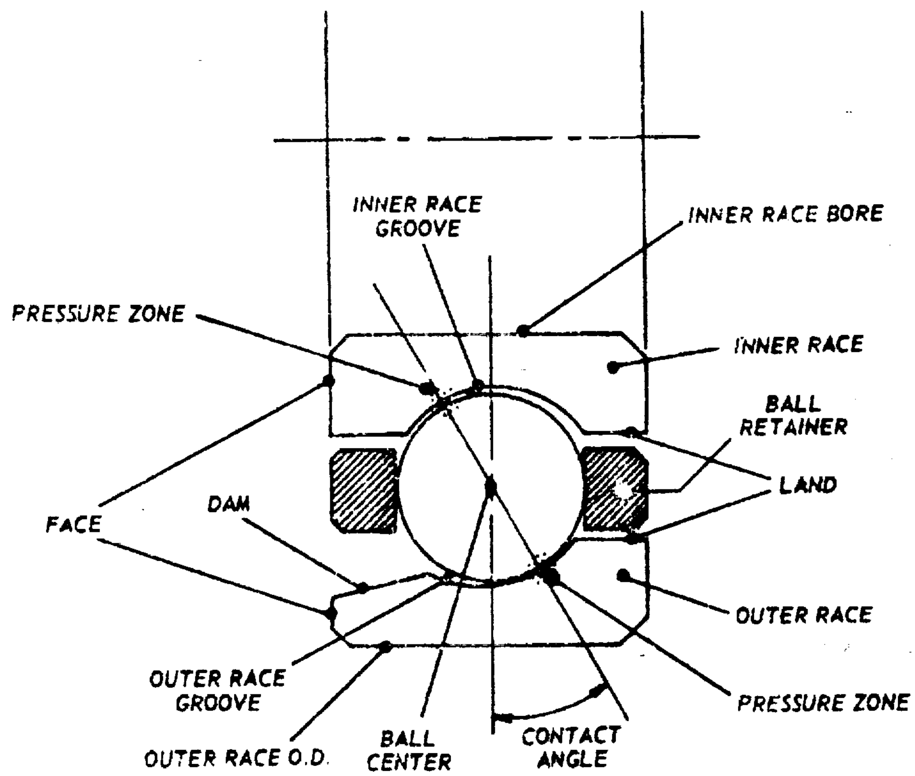
The aforementioned factors must be accompanied by very careful initial assembly and manufacture.

Proper controls and processing are of utmost importance for long life performance to prevent particulate contamination, chemical contamination, overheating, mounting distortion, overstressing, scratching, denting, over-lubrication, under-lubrication, exposure to corrosive environments and etc. Reference is made to ALERT-MSFC-69-5 (reproduced in Section I) where excessive heat exposure during gyroscope assembly was evidenced. With proper design and the aforementioned attention to the manufacturing stage of assembly, the bearing testing for incipients can be a reliable process for life determination.

A discussion of the various test methods that can be employed is covered later in this section; however, it is important to keep in mind that the gyroscope itself, its function and purpose, makes it inherently a very sensitive diagnostic device for bearing failure. As stated previously, the drift of the gyro increases and becomes more unstable as compared to previous test results as the bearing degrades.

Figure 2 shows a cross section of the ball-bearing to present the bearing nomenclature.

Much of the detail of each bearing application is coupled with the specific gyroscope configuration and requirement. There are, however, some points where there is greater commonality and the failure mode analysis will concentrate on these factors. Individual bearing configurations may vary with respect to the basic size, material, mounting method, preload, speed lubrication, contact angle, race-groove curvature, inner or outer land relief, retainer configuration and tolerances assigned to these factors. For example, if there is a requirement for performance under acceleration, the most critical bearing parameters may be the contact angle, number of balls, race-groove curvature and preload. Proper initial design is probably the most critical feature of long life performance.



$$\text{RACE GROOVE CURVATURE} = \frac{\text{RACE GROOVE RADIUS}}{\text{BALL DIAMETER}} \times 100\%$$

Figure 2 Bearing Nomenclature

2) *Detailed Ball Bearings Life Considerations* - Metals, geometry, surface and chemistry considerations are discussed in the following paragraphs.

Metal - The demands on the steel in the bearing construction is different in gyroscope bearings. The maximum stress is generally around 200,000 psi. The bearing generally operates in a temperature environment of about 150°F, and in an inert atmosphere. The demands upon the bearing metal evolves the basic requirement that the material must be free from elastrohydrodynamic film piercing asperities. Therefore, the race-groove and ball surfaces microstructure must physically and chemically support the EHD lubrication. Microdimensional stability must be maintained under the stresses and during the storage conditions that may vary from -85°F to 225°F.

In the various types of steels employed, factors during the initial process control include the chemical composition, the microstructure, the carbide, composition, and response to heat treatment.

Inclusions in the material create film-piercing asperities and lead to the breakdown the EHD film. The state-of-the-art in the steels used in bearing material is fully adequate for the bearing performance when rather strict adherence to process control is implemented. It should not be a major failure component in the long life performance of the bearing. Again, it should be stated that *material fatigue is not a primary initial failure mode* in gyroscope bearings because gyroscope bearings are not highly stressed.

Geometry - The geometry is associated with stress levels, EHD film thickness, and lubricant control. Also affected is the response to accelerations and producibility. All the tolerances of the bearing construction must be of proper design and control to maintain the proper pressure zone in the EHD film. The bearing technology is developed sufficiently so that runout and other tolerances can be measured to the required 5 to 50 microinch region. With the application of the *proper design and process control, runout and other tolerances should not be a life limiting factor.*

Surface - The primary failure involves the film-piercing asperities and the loss of the chemical support for the EHD film. The surface technology involving finishes from 0.3 to 3 microinches are generally sufficient for the EHD film support. The primary concern is for a surface to "wet" properly to maintain the 5 to 15 microinch EHD film. Again, this factor is not deemed to be a major contributor to premature failure in long life performance.

Chemistry - It is known that surface poisoning can take place and poor wetting can cause premature failure and excessive low speed torques. There are process techniques to correct and insure that there is proper "wetting". When properly controlled, these processes provide no limitation to long life performance. Some experience on the Skylab ATM Control Moment Gyros has indicated the difference between a "good" lot of bearings and a "bad" lot was the surface finish of the balls in the bearing that pretended proper wetting. See Section H of this chapter for a digest of previous experience.

Lubrication - Lubrication along with the geometry of the bearing are the most significant long life features. Lubrication is simply the maintenance of the EHD film by the retainer delivering to the balls, in a stable manner, the required amount of a lubricant with the needed properties. Stability in sensing gyroscopes requires the lubricant be controlled to prevent excessive local oil building that causes fill-thickness changes. Then there are two limitations:

1) Stability dictates the total amount of oil that can be carried in the lubrication system, and;

2) Sufficient oil must be available for long life and maintenance of a low-friction coupling between balls and retainer.

Sensing gyroscopes with ball bearings use oil contained in a porous plastic ball retainer. (Control Moment Gyroscope ball bearings may use a configuration with the oil reservoir in the retainer, external fed, or grease.) Oil used for years has been an oil formulated with an anti-oxidant and an anti-foam and a lubricity additive (e.g., Humble's Teresso V-78).

Retainer - For years a phenolic laminate has been used for the retainer, however, problems with the tubes or rods from which phenolic separators are manufactured has been encountered from piece to piece and in a single length so that the structure of the material was not dependable. The inconsistent chemical and physical properties along with the fact that the phenolic retainer holds most of its lubricant close to the surface (5% by weight) indicates poor general characteristics. Phenolic should not be used in long life applications. The retainer material proven satisfactory in gyroscope bearings is the Nylasant with a thorough porous structure with about 25% porosity. Although Nylasant is somewhat more difficult to manufacture, some gyroscope bearings have shown life over 30,000 hours.

Some work reported by Freeman (Reference 3) indicates that for a specific set of running conditions for R4 bearing, 12,000 rpm life with Nylasant retainers exceeds 20,000 hours. With laminated phenolic, it ranges from 5000 to 15,000 hours. With solid non-porous nylon that has oil retention nearly equal to that of the phenolic, it approximates 2000 hours. Doubling the speed to 24,000 rpm leaves Nylasant life essentially unchanged, reduces phenolic life to about 500 to 2000 hours and drastically cuts life of solid nylon to less than 24 hours. The Nylasant retainer performs well; however, like all other features of the gyroscope, process and other controls are required.

b. Gas Gyro Wheel Bearings - There are several similarities between the ball bearing and the hydrostatic (self pressure) gas bearing. These are:

1) Gas as a fluid forms a pressure medium, and;

2) Gas is subject to contamination.

For gas bearings, deterioration may consist of one or more of the following:

1) Deposit of contaminants on working surfaces;

2) Loss of lubrication, and;

3) Wear of operating surfaces.

The performance of gas-bearing wheels should be much more predictable than ball-bearing wheels because of simple geometry and dimensional precision. Experience indicates that failure eventually results from:

1) Progressive deterioration of the surfaces caused by the stop-start cycle, and;

2) Possible contamination and aging which is probably chemical in nature.

As the surfaces deteriorate, the starting torque level increases until the available motor torque will no longer start the wheel. In the absence of high-speed touch-downs (as may result from excessive slew rates) which could result in catastrophic failure, the expected system failure mode is a non-start. With cleanliness assumed, and with wear limited to less than the small amount required to precipitate a non-start, the gas bearing wheel provides uniform operating characteristics during its entire life. The gas bearing provides improved statility of the gyroscope performance and has the added advantages of having no ball and retainer dynamics. Life expectancy of 10,000 hours can be expected. Improvements in the choice of materials, cleanliness, finish and lubricants may be expected to increase the life expectancy and decrease the unit-to-unit life-expectancy dispersion. Present technology, however, cannot predict either long or short life for a given gas bearing.

As with the ball bearing, the rate of deterioration is affected by the operating temperature. The failure to start due to the deterioration of the surfaces is alleviated by the choice of the bearing surface material - materials such as ceramics, flame plate aluminum oxide, tungsten carbine, chrome oxide, or boron carbide. The choice of material is governed by the requirements to provide a consistent and low starting friction for the bearing surfaces and to survive the required number of starts and stops.

The basic design and design application is associated with performance over the long period of time. Different stabilities in gas bearings as opposed to ball bearings influence the operation of the bearings and is a consideration in the application of gas bearings.

1) Half Speed Whirl - This occurs on the rotating journal bearings and manifests itself as either or both an orbiting of the journal center of mass or a coning of the journal axis at a frequency very nearly one-half of the rotor speed. Also, if the bearing sleeves are resiliently mounted, they may also exhibit half-frequency whirl. The half-frequency whirl is a self excited oscillation. An unloaded journal bearing is unstable. However, it can be made stable by operating at sufficiently high eccentricity ratios. High eccentricity ratios is a design derived parameter.

2) Synchronous Whirl - This occurs due to forced vibrations. It is simply the natural tendency of any resilient mechanical system to respond to vibratory inputs. Rotor unbalance is one source of the vibration input and causes the bearing system to respond at the same frequency as that of the input--1 cycle per revolution--thus "synchronous".

To prevent the half-frequency whirl, design to either avoid the low eccentricity ratios or add steps or grooves to the surface similar to those used in the thrust bearings. The design to prevent synchronous whirl is proper choice of materials and precision balancing of rotating elements.

Figure 3 sets forth the types of gas bearings and Table 2 compares the gas bearing configurations. Simplicity of achieving the manufacture and design goes hand in hand with the ability to successfully apply process controls for a successful long life design.

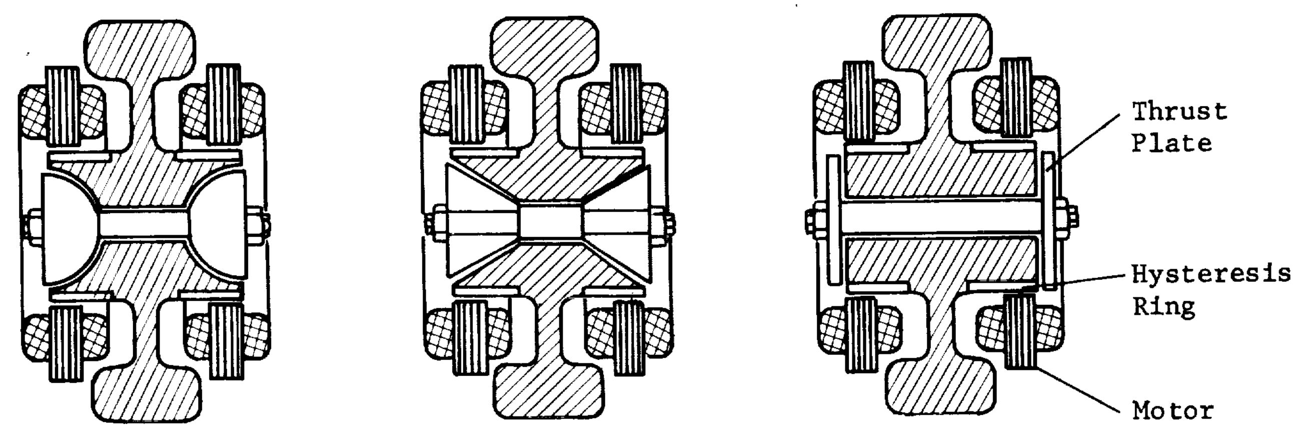


Figure 3 Types of Gas Bearings

Table 2 Three Types of Bearing Configuration

Type	Angular Tolerances and Alignment of Axes	Dimensional Tolerances	Spiral Groove Manufacture for Thrust Support	Mutilation for Whirl Suppression	Suitability for Coating Surfaces with Hard Materials
"H" Type	4 thrust surfaces to be square to the journal axes to 3 arc seconds Very difficult	Diameter of journal or bearing (10 microinches) Length of journal or bearing (10 microinches)	Straightforward and relatively cheap methods available	Lobing usual but requires much hand-work and inter-mediate inspection Spiral grooves more effective but also difficult to make	Very difficult to coat bores
Opposed Hemispheres	None	Diameter of either male or female hemispheres Difficult metrology or manufacturing problem	Very difficult and probably expensive Pockets much cheaper but not so effective	Not required	Relatively easy to coat both surfaces
Conical	Cone angles to be the same Relatively easy Axes of the 4 bear-ing surfaces to be accurately alligned Difficult	None (except of a separately manu-factured spacer)	Almost as easy as with flat thrust plates (for cones of semi-angle 30° or more)	Not required	Very easy to coat both surfaces (for cones of semi-angle 30° or more)

Some examples of solutions to contamination in gas bearings is contained in Reference 4.

Experience in the preliminary build process of GG159E for a sterilizable gyroscope indicates:

1) The lockup of specific gas bearing motors was due to organic-particulate contamination and not to vapor-condensed materials.

2) The contamination was associated with bonding and encapsulating materials in the motor stator.

3) Materials used in spinmotor fabrication compare favorably with alternate materials tested either as used or with some additional processings. No material changes were implemented.

4) Water vapor and organic outgassing tests verified the original assumption that these phenomena would be much more prevalent at sterilization temperature. As a result, special vacuum-bake processes were initiated as a prevention.

This seems to illustrate the sensitivity of the gas bearing to contamination, although the sterilization temperature exceeds the temperature of gyroscopes used in ordinary manned spacecraft application. This also suggests high temperature bake may enhance long life by eliminating the organic contaminants in a manufacturing process.

c. *Materials* - Encapsulating materials can change characteristics with time and effect scale factor and bias torque levels through stator permeability changes. Differential changes (around the stator poles) cause bias shifts. The possibility that aging of stator lamination-bonding materials could affect the stator permeability has not yet been established as fact.

2. Design

a. *Selection Criteria* - The following is the general selection criteria for gyroscopes. Again it is important to stress that each type of model number of the gyroscope is unique, and there is little commonality between the different manufacturers. Therefore, the selection choice is limited unless a new development is required, and in that case, the experience factor for long-life design may be lacking entirely.

1) Motor Bearing Assembly - The gyroscope application should be closely examined to determine whether the proper choice for the wheel rotational interface should be ball bearing or gas bearing. The decision to select a ball or gas bearing wheel for a given mission is a difficult one to make. One criterion for making the choice is to evaluate the history of proven bearings in view of system requirements, with respect to life expectancy, performance, and design environment.

Pertinent parameters include:

1) Power required (temperature related) and its stability;
2) Starting/running torque;
3) Voltage (an electrical noise source);
4) Wheel assembly motion perturbations (mechanical noise sources);
5) Anisoelasticity;
6) Life restart reliability;
7) Mechanical environments (linear and angular rates);
8) Acceleration;
9) Vibration and shock (inside and outside the system);
10) Run-up time;
11) Lubricant;
12) Running life;
13) Transmassibility;
14) Shelf life;
15) Manufacturability;
16) Spin motor reaction torque;
17) Wheel speed.

The following items must be considered for ball bearing usage:

1) Wheel mass stability required;

2) Bearing stiffness (contact angle, preload);

3) Mechanical environments;

4) Start and run torque margins;

5) Lubricant compatability with operating temperature range;

6) Required stability of operating temperature;

7) Run-up/run-down time;

8) Motor running power;

9) Thermal resistance to the control point or heat sink;

10) Ball bearing and retainer dynamic frequencies and their amplitudes;

11) Wheel synchronization monitor;

12) Torque margins.

Gas bearings should be specified for use if highest performance is required, and if the ball and retainer dynamics of the ball bearing are intolerable. The following parameters must be considered for gas bearings:

1) Wheel mass stability;

2) Bearing stiffness (gas pressures and running support forces);

3) Mechanical environments (especially those such as slow rate, which can cause bottoming of the wheel at speed);

4) Bearing type and materials, including lubricant thermal characteristics;

5) Start/stop life expectancy;

6) Torque margins;

7) Motor power stability;

8) Run-up, run-down times;

9) Motor running power;

10) Permissible operating temperature range;

11) Required operating temperature stability;

12) Thermal resistance to the control point or heat sink;

13) Wheel synchronization monitor.

The choice between the gas bearing gyroscope and the ball bearing gyroscope could be ranked as following:

First Choice - If there is an identical application, identical specifications and interfaces required in the new vehicle application to that of an existing application and the existing application has had a history of successful long-life, then the choice of that same gyroscope for the new application should be made.

Second Choice - If the application has no starts and stops in the gyroscope wheels, and can furnish continuous power, then the choice should be for a gas bearing gyroscope. Caution should be made to careful manufacturing controls as previously set forth in this report for contaminants can cause immediate seizure of the bearings.

Third Choice - If the application requires starts and stops of the gyroscope wheel, use a ball bearing gyroscope. Periodic testing for lubrication film stability will probably be required (shows up as excessive drift).

Fourth Choice - If the application requires a low noise gyroscope, then the gas bearing gyroscope is the choice in any case because the ball bearing gyro possesses more output noise due to the balls and the retainer.

2) Other Selection Criteria Factors - Repeated temperature cycling of a gyroscope prior to acceptance testing, through a range representative of the expected mission, will speed up aging and improve drift stability.

Specify treatment of "shelved" or "stored" gyroscopes with regard for mechanical, thermal and magnetic environments.

Float structure should be as symmetrical and rigid as practicable. The thermal conductance of the float assembly should be consistent with the performer constraint considering wheel power, bearing temperature, thermal characteristics of the damping fluid and damping gap.

A method to reduce wheel power by techniques of periodic overexcitation could possibly reduce the heat in the wheel drive motor. Other wheel power requirements are power margins, the heat conduction characteristics of the float, gap, and fluid.

b. Results of Survey - A survey was conducted on those gyroscope families which had large production runs. Much reliability data exists to document the conclusions set forth in Table 3. In all cases of successful usage, a long design-development program existed prior to contractual commitment to purchase production quantities.

Further, consistent in the survey is the opinion that the design must be conservative. Gyro performance specifications exceed the system requirements. The greater margin enchances reliability.

Testing, during and after the manufacturing phase, plays an important role in screening would be failures at a premature date. The would be failures can be reduced by intensive inspection, material lot control, and establishment of exacting fabrication assembly and testing procedures. There is, however, the principle of diminishing returns involved; and the industry in general feels that they are operating near optimum.

However, gyroscopes used in missile and satellite applications require much more control over build and test than do gyroscopes used for atmospheric, surface and marine applications, primarily because of the consequences of such a failure in a missile application.

Table 3 Survey of Users and Manufacturers on Specific Parts

Part Description and Using Program	Users Opinion of Why Part Was Successful	Manufacturer's Opinion of Why Successful	Recommended Guidelines for Improvements, Procurement Specs, Requirements	Relative Impact on Yield, Cost of Incorporating Recommendations
Teledyne 800-5001 Gyroscope (Classified)	Intensive screening program initiated to verify performance.	Design specifications far exceeded the requirements.	Raise momentum of gyro to get better performance	Yield increase by factor of 2.
Kearfott Gyroflex Gyro A-7 & SRAM (KT-70 Series)	High volume production runs. Practices and procedures well established.	Long developmental and testing program, manufacture techniques.	Recommended correct interpretation of component specs when comparing with system requirements. Install heat pipe cover.	$500/gyro.
Nortronics GIT-1 B Minuteman	Good conservative design, specs exceed system requirements.	Simultaneous test and development program, materials control.	Spec out parameters required by a specific application. Shopping list purchase of inertial components - dangerous.	
Kearfott TEC-C	Is not operational yet, but is surviving very severe test program.	Much experience with gyro design, manufacturing techniques, lot control of piece parts.	Caution against exceeding design specs in any given application.	
Autonetics G-9 Minuteman	USAF initiated an intensive reliability program to assure success.	Numerous design changes during development and test phase extensive testing.	None	
Litton G300-G2, F-111	Extensive development and test program.		Use trained personnel in handling and test.	Replacement and repair cost are reduced.
MIT 25PG10-H	Screening and run-in procedure during motor buildup and preload phase.	500 starts/stops under test during build phase.	Recommended running acceptance test and monitor for many hours to detect trend. 30,000 hours.	Cost of testing not prohibitive.
Litton LN-30 Vibragimbal, F-15	Six-year development program.	Good basic design and much in process control.	None	
Singer-Kearfott Alpha-2 Titan	Design specs conservative, good in-process control. 100% receiving inspection at customer facility.	Stringent process controls, good design.	None - no flight failures in 800 units. Recommend project engineer for each project.	None

c. Design Alternates - The grease bearing possesses a potential for long-life. Recent activity (Reference 1) indicates that bearings fitted with spiral grooves and lubricated with grease will develop a full fluid film, provide high load capacity, excellent tolerance to overload and start-stop rubbing and have the potential for a long operating life, provided: (1) the bearing design is such that the net lubricant flowout of the clearance space is zero, and (2) the lubricant physical and chemical properties remain relatively constant over the operating life.

The two requirements above apply to both bearings in small gyros or large momentum wheels and control movement gyros.

Grease has been studied and found to have acceptable power losses and mechanical characteristics at speeds as high as 24,000 rpm's.

Compared to a gas lubricated bearing, a grease lubricated bearing has the following general advantages and disadvantages:

Advantages:

1) Higher load capacity to power loss ratio;

2) Better high speed rub and stop-start characteristics;

3) Greater tolerance to dirt; component handling and clean room assembly conditions less critical;

4) Greater safety margin for overload;

5) Lower rotor windage losses due to ability of bearing to operate in low ambient pressure;

6) Cheaper to make because of less stringent manufacturing tolerances accrued from greater operating film thickness;

7) Less stringent surface finish and material dimensional stability; bearing surface porosity or accidental scratches produce less of a change in performance characteristics.

Disadvantages:

1) Possibility of loss of lubricant or vaporization of fluid in lubricant;

2) Possibility of cavitation and gas entrainment;

3) Possibility of alteration in lubricant properties due to chemical and temperature changes;

4) Change in rotor angular momentum due to lubricant mass shift.

Compared to ball bearings, a grease lubricated bearing has the following general advantages and disadvantages:

Advantages:

1) Longer life at higher rotor speeds;

2) Less noise and vibration.

Disadvantages:

1) Not "off-the-shelf" item (currently) and must be designed rather than selected from a manufacturer's catalog;

2) Less overload safety margin.

d. Hardware Life - Gyroscope life has been discussed in the failure mechanism analysis section. Backup life data is presented in the previous experience section H. Depending upon application, the gyroscope life is estimated to range from 3000 to 10,000 hours. Life is limited by the bearing life.

e. Application Guidelines - These guidelines are discussed throughout this chapter. In summary, be certain that the gyroscope specifications meet or exceed the application requirements for all applications. The most critical parameters pertaining to reliability are vibration, temperature, shock and acceleration.

D. TEST METHODOLOGY AND REQUIREMENTS

With the diverse application and large number of different types of gyroscopes, specific test methods applicable across the range of parameters is not appropriate . The gyroscope drift trends over a period of time is a very sensitive indicator of degradation in stability over long time. Drift tests are generally well established and documented in the literature.

Performance bands placed on the results of tests determine if a particular gyroscope failed to have proper performance over its lifetime. It indeed determines life limits for a particular gyroscope design. For example, gyroscope applications with wide performance bands will operate over a longer period of time than those with more stringent performance requirements.

The following paragraphs present examples of some of the test techniques that can be applied to enhance long-life.

1. Optical Test Process

In the preassembled condition, optical inspection tools include microscopic inspection, polarized light, interference microsope to detect topographic operations. Other tools are readily available for measuring geometry and surface finishes. Most of the optical inspection processes are standard operating procedures at the bearing supplier or the gyro manufacturer's location.

2. Low Speed Dynamometer

A very sensitive technique employed at the motor assembly state of manufacturing is the low speed dynamometer. This is used for testing the assembled bearing. This test consists of a spindle, a dead weight axial loading system and a bearing whose inner face is mounted so that the strain can be measured between the inner race and the support structure. Note: On almost all gyroscope wheel bearings, the outer race is attached to the wheel and the inner race is to a shaft fixed to the gimbal. The low speed dynamometer gives an indication of the assembled bearing surface and the lubrication characteristics between, as on ball and race, as an output from the strain race measurement.

in the low speed dynamometer, the direction of rotation, load, speed, and zero setting are variable. The strain measurement is a torque reading in which the torque level, the material surface, the lubricant conditions, the contamination and the geometry is indicated.

3. Milliwattmeter

On the completed gyro assembly, and on completed gyro package units where provision is provided, the milliwattmeter test can be employed. This is simply a sensitive zero suppressed power input measurement. Changes in the power trace reflect changes in the bearing torque that in turn are indication of variations of the gyro axial positions.

4. Deceleration Tests

This common test consists of determining the time from power removal to zero wheel speed. The time versus speed history at the high speed segment yields bearing friction and windage data. The low speed segment is not influenced by windage. To enable this test to be conducted at the systems level, gyros should incorporate spin motor rotation detectors (SMRD) and a means of electrically decoupling the gyros since the generation of back e.m.f. materially effects the run-down time.

Gyroscope performance stability is the most sensitive method of determining the performance. As stated before, the slight deviation of the EHD film results in a detectable change in the characteristics in a one g force field--earth gravitational field. For example, a fraction of a microinch out of a total film thickness of 5 microinches (typical) can show a deviation in the drift stability of one meru for 0.015 degree per hour. Gyro performance tests for single degree of freedom gyroscopes include:

1) Tumble test - a rotation of the gyroscope about the output axis until the input axis is in a place normal to the earth rotational axis. This provides a force of gravity in the float, and by recording the feedback current and indication of the mass unbalance caused by the breakdown of the EHD film can be detected.

2) Fixed position tests - fixed position tests instead of continuous tumbling can provide some of the data on the float balance/bearing conditions.

E. PROCESS CONTROL REQUIREMENTS

Process control reduces itself to the most important feature of the gyroscope. Contamination is a critical factor because a fractional microinch EHD film must be maintained. Chemical, particulate and other contamination must be avoided.

There are several testing techniques on the various levels of assembly. Figure 4 shows the various tests that may be employed during the assembly operation. Of the six different tests, four can be performed as a final test along with the final test.

Test Process Techniques	Single Bearing	Bearing Pair Test	Wheel	Gyro
Optical	●	●		
Low-Speed Dynamometer	●	●	●	
Milliwattmeter		●	●	●
Deceleration Test		●	●	●
Frequency Analysis		●	●	●
Gyro Test				●

Figure 4 Test Processes

The following are "title" descriptions of specification features relating to long-life in the Mariner Gyro Specifications:

1) Materials, Parts, and Processes - calls for screening and traceability.

2) Spin Motor Jag Requirements - calls for noise limits at hunt frequency.

3) Spin Motor Characteristics - calls for maximum run-up time, minimum run-down time. Bearing preload, and maximum run-up without heaters.

4) Tolerances are Pyramided - more stringent at vendor final test than at customer's location.

5) Internal Contamination - calls out clean room particle count.

6) Vacuum Temperature Cycling.

7) Lot Control on Drawing Changes, Design, Material, and Processes.

8) Traveler - calls out designation of individuals assigned to each operation. This includes designation of certain assembly operations to be done by same individual.

9) Lot Sampling Requirements - calls out quanity and inspections to be performed at specified stages of assembly.

10) Selection/Process for Spin Motor.

11) Run-in Tests over Specified Period of Time.

F. PARTS LIST

Table 4 presents a list of gyroscopes used in aerospace applications. Also delineated are the design and operational characteristics. A gyroscope satisfactory in one application may not be satisfactory for another application. Hence, it is difficult to state that some are acceptable and others not acceptable.

Table 4 Gyroscope Part List

Gyro Manufacturer	Gyro Model No.	Type	Rotor Bearing Type		Angular Momentum (CGS)	Gimbal Freedom	Gyro Gain	Pickoff (Type)	Torque (Type)
Nortronics	G1-M1	SDF	Gas	Flotation electron static	2.4×10^5	±3 deg max	15	Capacitive	Perm magnet
Nortronics	G1-K7	SDF	Ball	Flotation pivot and jewel	0.9×10^5	±3 deg	1.0	Microsyn	Perm magnet
Nortronics	GRH4	SDF Rate							
Kearfott	2542 2543	SDF	Ball	Flotation, pivot and jewel	0.3×10^5	±2 deg	1.0	U-bridge air core	Perm magnet
Kearfott	2564	SDF	Ball	Flotation, pivot and jewel	2.27×10^5	±3 deg	0.3 to 250	U-bridge air core	Perm magnet
Kearfott	2590	SDF	Gas	Flotation, pivot and jewel	3.5×10^5	±2 deg	8.8	Microsyn	Perm magnet
Kearfott	2566	SDF	Ball	Flotation, pivot and jewel	2.27×10^5	±2 deg	1.0	Microsyn	Perm magnet
Kearfott	2536	SDF	Ball						
Kearfott	2519	SDF/RD							
Kearfott	2514	SDF/RF							
Kearfott	2565	SDF/RI							
Kearfott	2021	SDF Rate							

r	Motor Volts	Freq (cps)	Power (w)	Range	Usage	Remarks
	26	3ϕ 1600	6.5 Start 3 Run	0-1000 deg/hr (0.278 deg/sec) 0-6300 deg/hr available (1.75 deg/hr)	Company development	Very early development, ceramic considerations, torquing rate is too low for strap-down usage.
	26	3ϕ 400	3.26 Start 3.0 Run	17 deg/sec cont 60,000 deg/hr	Vela, Lorv, Flip, Arpat, OSO, SAS-D	Three units modified for 30 deg/sec continuous torquing arc being fabricated for TRW Systems.
					Saturn, ERTS, Nimbus	
	26	3ϕ 800	2	20 deg/sec cont 30 int 10 deg/sec cont 20 int	Company development	2543 torquing rate is too low for strap-down usage. 2542 is "similar" in price to Nortronics G1-K7, but has poorer drift stability and no production history.
	26	3ϕ 400	4.5 Start 3.5 Run	20 deg/sec cont	Rags, Tars, Mariner C, Agena, ERTS, Lunar Orbiter, Nimbus, OAO	This unit is similar in price and performance to the Honeywell GG334A, but it was designed for platform usage and does not have the high torquing rate capability of the 334A.
	26	3ϕ	15 Start 8 Run	0-11,000 deg/hr _____		Torquing rate is too low for strap-down usage.
	26	3ϕ	3.7 Start 2 Run	25 deg/sec cont	Proposed for IM abort guidance system	High torque modification of 2565 Alpha Series Gyro. However, this is a conceptual design and no hardware development is planned.
					Titan	
					Skylab (ATM)	
					Surveyer	
					Viking Orbiter Mariner	
					LEM	

Table 4 (cont)

Gyro Manufacturer	Gyro Model No.	Type	Rotor Bearing Type		Angular Momentum (CGS)	Gimbal Freedom	Gyro Gain	Pickoff (Type)	Torq (Typ
MIT (Design)	251RJG Mod II	SDF	Ball	Flotation, magnetic	1.3×10^5	±1 deg	1.0	Micro-syn	Micr syn
Sperry	SYG 1000	SDF	Ball	Flotation, pivot and jewel	1×10^5	±4 deg	0.5 to 14	Moving coil	Perm magn

…er …)	Motor Volts	Freq (cps)	Power (w)	Range	Usage	Remarks
–	28	2φ 800	8 Start 4.5 Run	25 deg/sec max	Polaris system, OSO, Apollo, Apollo LM	A modified version with a permanent magnet torquer is used in the MIT prototype system, constant temperature control (even during transportation) is required. (Designated 25 PIRIG)
…et	26	3φ 400	4.2 Start 3.0 Run	0-28,000 deg/hr cont 67,000 deg/hr (8 scc max)	Lunar Orbiter	Torquing rate is too low for strap-down usage. Modified version with air bearing and high torquer rate (~100 deg/sec) desgnated SYG 1440.

Table 4 (concl)

Gyro Manufacturer	Gyro Model No.	Type	Rotor Bearing Type		Angular Momentum (CGS)	Gimbal Freedom	Gyro Gain	Pickoff (Type)	Tor (Ty
Honeywell	GG49	SDF/RI							
Honeywell	GG53	SDF/FRE							
Honeywell	GG87 GG76	SDF	Ball	Flotation, pivot and jewel	___ x 10^4	4 deg	0.3	Dualsyn moving iron	Per mag
Honeywell	GG79	SDF/ Rate							
Honeywell	GG327	SDF	Gas	Flotation pivot and jewel	2.0 x 10^5	Up to ±4 deg	0.4	Micro-syn	Per mag
Honeywell	GG334A	SDF	Gas	Flotation, pivot and jewel	___ x 10^5	±3 deg	0.5	Micro-syn	Per mag
Honeywell	GG49								
Honeywell	JRT45/ Rate	SDF/ Rate							
Honeywell	MS133	SDF/ Rate							
Honeywell	MS100	SDF/ Rate							
Honeywell	MS130	SDF/ Rate							
MIT 2FBG-2C		SDF/RI							
NASA/Bendix	AB-5	SDF	Ball	Hydrostatic bas gearing	2.6 x 10^6	±3 deg	Very large	Moving coil	Mo
Hamilton Standard	1139B 1139E	SDF	Ball	Flotation, pivot and jewel	___ x 10^5	±1 deg	0.25	Moving coil	Per mag
Hamilton Standard	1139S	SDF/RI							

	Motor					
er	Volts	Freq (cps)	Power (w)	Range	Usage	Remarks
					OGO	
					Mercury	
	26	3ϕ 400		0-60 deg/sec 0-30,000 deg/hr	Scout, PRIME, Agena B	Superceded by GG334.
					Mercury	
ı gt	50 Start 26 Run	3ϕ 800	4.5	0-180,000 deg/hr 50 deg/sec 0-1000 deg/hr (0.278 deg/sec)		Superseded by GG334.
n gt	26	2ϕ 800	7.0 Start 3.5 Run	114 deg/sec cont	Honeywell strap-down system Agena	This unit was specifically designed for strap-down usage.
					Burner 2, Centaur, Ranger	
					Biosatellite Explorer-31	
					Gemini	
					Mercury	
					OGO	
					Titan 1G, OAO	
ig	26	3ϕ 400	8	0-360 deg/hr	Saturn IC system	From "A General Description of the ST 124-M Inertial Platform System NASA TN D2983," Sept. 1965, H. E. Thomason.
n gnt	26	3ϕ 100	6.0 Start 3.0 Run	28 deg/sec cont	AGS, LM abort system, Delta	The high cost of the unit and its low torquing rate in comparison to the 334A make it a less attractive unit.
					Viking	

G. GENERAL DESCRIPTION

Gyroscopes have been used on a variety of space vehicles to provide either information on the attitude or angular velocity of the vehicle or as an inertial platform with respect to a reference coordinate system. Tables 5a and 5b set forth gyroscope applications and characteristics. Table 6 presents typical dynamic ranges of gyroscopes.
The gyroscopes are key elements. The performance of the associated space vehicle system is strongly dependent upon the performance of the gyroscopes. It is therefore essential that the gyroscope package be designed consistent with the mission requirements in order to provide the required information accuracy and reliability.

Major factors which influence the design of gyroscopes for space vehicle systems include:

1) Configuration (strap-down or stable platform mounted);

2) Performance (accuracy, drift rate, resolution, dynamics, etc.);

3) Reliability (MTBF, redundancy, etc.);

4) Lifetime (shelf life, operating life, testing duration, duty cycle, etc.);

5) Environment (acceleration, vibration, thermal, magnetic, radiation);

6) Cost (money, weight, volume, power, time), and;

7) History (experience of gyro and manufacturer, production status, etc.).

Gyroscope technology and the application of gyroscopes to space vehicle systems have advanced rapidly in the past 15 years. The number of gyroscopes required for most space vehicle applications is usually small; consequently, most of the gyroscopes used in space vehicles have been adapted from gyroscopes developed for other applications, such as military.

Table 5a Gyroscope Applications

Mission	Gyro		Application	
Agena B	Honeywell GG76	SDF-RI	Guidance, Flight control	Strapdown (3)
	2 Honeywell GG76	SDF-RI	Guidance, Flight control	Strapdown (3)
	1 Honeywell GG87	SKF-RI		
Agena	Kearfott 2564	SDF-RI	Guidance, Flight control	Strapdown (3)
	Honeywell GG-334	SDF-RI	Navigation, Guidance, Flight control	Strapdown (3)
Atlas	1 Kearfott 2506	SDF-RI SDF-rate	Flight control, Stabilization	Strapdown Strapdown
Burner 2	Honeywell GG-49	SDF-RI		
Centaur	Honeywell GG-49 D26	SDF-RI	Guidance, Attitude reference	Platform (3)
Delta	Hamilton Std. RI-1139E	SDF-RI		Strapdown (3)
Saturn 1B	Bendix AB-5-K4 Nortronics GRH4T	SDF-DRI SDF-rate	Guidance, Stabilization	Platform (3) Strapdown (9)
Saturn 5	Nortronics GRH4T Bendix AB-5-K8	SDF-rate SDF-DRI	Stabilization, Guidance, Navigation	Strapdown (9) Platform (3)
Scout	Honeywell GG87	SDF-RI SDF-rate	Autopilot Autopilot	Strapdown (3) Strapdown (3)
Titan IIIA	MIT 2FBG-2C	SDF-RI	Guidance, Navigation	Platform (3)
Titan IIIC Titan IIIC	MIT 2 FBG-2C MIT 2 FBG-2C	SDF-RI SDF-RI	Guidance, Navigation	Platform (3)
*	DELCO 651G	SDF-RI	Guidance, Navigation	Platform (3)
	Kearfott 2586	SDF-RI	Stabilization	Strapdown (3)
Titan Tran-stage	Kearfott 2586	SDF-RI	Stabilization	Strapdown (3)

* Primary current use is aircraft navigation. It is programed for use in the Titan IIIC Standard Space Guidance IMU. This gyro has experienced more than 3.4×10^6 hours in the over 1000 Carousel IV systems built since 1968.

Table 5a (cont)

Mission	Gyro		Application	
OSO	Bendix 251RIG	SDF-RI	Stabilizing	Strapdown (1)
	Northrop GI-K7G	SDF-RI	Pointing	Strapdown (3)
Ranger (1 and 2) (Subsequent)	Honeywell GG 49-E5	SDF-RI	Stabilization	Strapdown (3)
	Honeywell GG 49-E12	SDF-RI	Stabilization, Attitude reference, Pointing	Strapdown (3)
SAS D	Northrop GI-K7G	SDF-RI	Pointing	Strapdown (6)
Skylab workshop	Kearfott 2519	SDF-RI	Pointing Stabilization	Strapdown (9)
Surveyor	Kearfott 2514	SDF-RI	Attitude reference, Stabilization	Strapdown (3)
Viking Lander	Hamilton Std. RI-1139S	SDF-RI	Inertial reference	Strapdown (4)
Viking Orbiter	Kearfott 2565	SDF-RI	Attitude reference, Stabilization, Pointing	Strapdown (6)
Apollo CSM (SCS)	MIT 25 IRIG	SDF-RI	Navigation, Stabilization	Platform (3)
	Hamilton Std. RI-1130	SDF-RI	Backup Stabilization	Strapdown (3)
Apollo 1M	AC 25 IRIG	SDF-RI	Navigation, Stabilization	Platform (3)
AGS	Hamilton Std. RI-1189B	SDF-RI	Backup attitude reference, Stabilization	Strapdown (3)
LEM	Kearfott 2021-B01	SDF-rate	Stabilization	Strapdown (3)
ATM	Kearfott 2519	SDF-RI	Pointing, Stabilization	Strapdown (4)
Bio-satellite	Honeywell JRT45	SDF-rate	Attitude reference, Stabilization	Strapdown
ERTS	Kearfott 2564	SDF-RI	Stabilization, Attitude reference,	Strapdown (1)
	Nortronics GRH4	SDF-rate	Initial stabilization	Strapdown (1)

Table 5a (cont)

Explorer 31	Honeywell JRT45	SDF-rate	Stabilization	Strapdown
Gemini	Honeywell MS-133 Honeywell GG-8001	SDF-rate SDF-RI	Stabilization Navigation	Strapdown (6) Platform (3)
Lunar Orbit	Sperry SYG-1000	SDF-RI	Attitude reference, Stabilization	Strapdown (3)
	Kearfott 2564	SDF-RI	Pointing (Backup)	Strapdown (3)
Mariner	Kearfott 2565	SDF-RI	Stabilization, Attitude reference, Pointing	Strapdown (3)
Mercury	Honeywell GG-79A Honeywell GG-53 Honeywell MS-100	SDF-rate 2DF-free SDF-rate	Stabilization, Attitude reference Attitude, Rate display	Strapdown (3) Strapdown (2) Strapdown (3)
Nimbus	Kearfott 2564	SDF-RI	Stabilization, Attitude reference,	Strapdown (1)
	Nortronics GR11	SDF-rate	Initial stabilization	Strapdown (1)
OAO	Kearfott 2564 MIT 2 FBG	SDF-rate SDF-RI	Stabilization Stabilization	Strapdown (3) Strapdown (3)
OGO	Honeywell MS 130B1	SDF-rate	Stabilization	Strapdown (1)
	Honeywell GG-49	SDF-RI	Pointing	Strapdown (2)

Table 5b Gyroscope Characteristics

Application	Gyro Types	Frequency Range	Torquing Rate	Drift Rate
Guidance and navigation	2DF SDF-RI	Platform, 0.001 to 1 Hz Strapdown, 0.001 to 10 Hz	Platform, 50 deg/hr or less Strapdown, up to maximum vehicle rate	0.05 deg or better
Stabilization and control	SDF-rate SDF-RI 2DF 2DF-free	0.1 to 10 Hz	Up to maximum vehicle rate	0.1 to 50 deg/hr
Tracking and pointing	SDF-rate 2DF-free	Up to 100 Hz	Up to 4000 deg/sec	0.1 to 10 deg/min

Legend:

2DF Two Degree of Freedom

SDF Single Degree of Freedom

RI Rate Integrating.

Table 6 Typical Dynamic Range of Gyros

Open-loop, spring-restrained gyro	8000:1
Ball-bearing suspension, open-loop gyro	20 000:1
Dry, servoed gyro; ball-bearing gimbal mounts with feedback through torquer	40 000:1
Floated gyro; non-temperature-controlled	100 000:1
Fully-floated, temperature-controlled gyro	600 000:1

Gyroscopes generally are: (1) a free (displacement) gyro measure rotation of the gyro case with respect to space by sensing the output angle(s) between the gyroscope element and the case, and (2) a "captured" gyroscope is operated so that the output angle between the gyro element and the gyro case is kept near null. The capture is accomplished either by torquing the gyro element so that it rotates with respect to space or by using a servo followup which causes the structure that supports the gyro case to rotate with respect to space. The information from a displacement gyroscope comes from an angle measurement, while the information from a captured gyroscope is the result of a torque measurement.

Two other gyroscopes now being applied are the free-rotor gyroscopes and the tuned rotor gyroscopes.

A free-rotor gyroscope is a 2DF gyroscope without gimbals. Other means such as flotation, gas bearings, electrostatic suspension or magnetic suspension are used to support the rotor. The gyro rotor is nearly spherical and is not mechanically restrained to rotate about a particular axis. All free-rotor gyroscopes must have some means of either preventing or damping nutation. Many gyroscopes of this type are kept running to minimize the number of starts and stops. Elements of SDF and 2DF gyroscopes are shown in Figures 5 and 6, respectively.

Another 2DF gyroscope uses a rotation flexure suspension on one end of the drive shaft to support the rotor. This all mechanical, nonfloated design is extremely simple. The decoupling of the rotor from its drive shaft and support bearing gives the gyroscope very stable g-sensitive drift and inertial quality of relatively low cost. Gyro performance failures due to spin axis mass unbalance shifts do not occur with microscopic shifts of the inner bearing race along the spin axis. However, the instrument is very sensitive to inputs at the *tuned frequency* or its harmonics.

The operating conditions may have an important influence on the gyroscope reliability and life. The following observations are typical:

1) Excessive acceleration, shock or vibration will reduce gyro life.

2) High temperatures or high-temperature gradients across the gyro can be deleterious to performance and mechanical integrity.

3) Human blunders occasionally cause gyro damage; e.g., spinning a gas bearing gyro backwards can ruin the bearing. A phase-order detection circuit improves reliability by protecting against such a possibility.

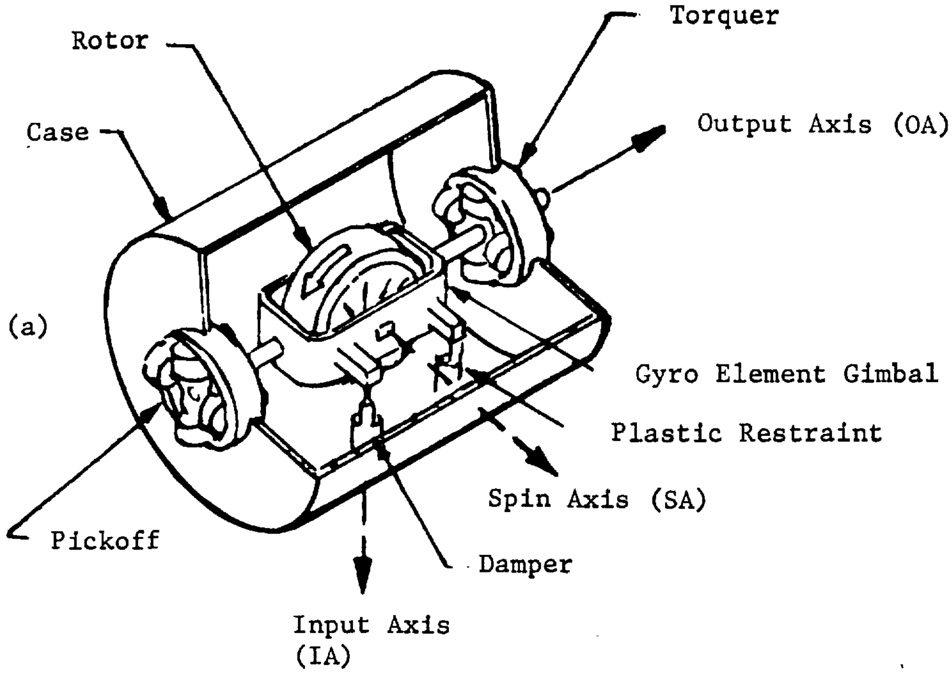


Fig 5 Essential Elements of SDF Gyro

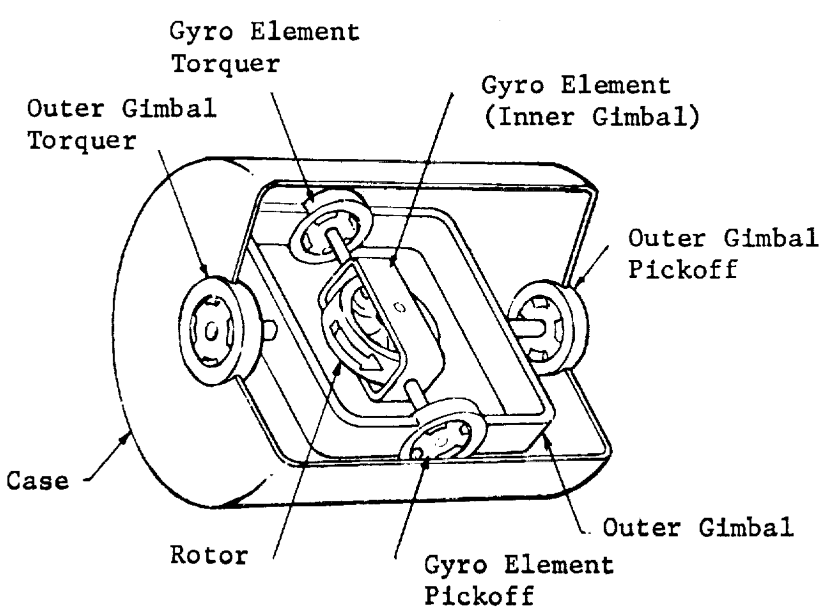


Fig 6 Essential Elements of a 2DF Gimballed Gyro

H. PREVIOUS EXPERIENCE

1. Apollo Experience

The experience with gyroscopes used on the Apollo guidance and navigation platform indicated that two types of production gyroscopes were used; one designated for Apollo I and the other for Apollo II, but used wheels of identical design. Analysis of data from each type showed considerable differences in both performance and life. The drift repeatability of Apollo II was better than that of Apollo I by a factor of two to one; however, the failure rate for the wheels of Apollo II was three times as high as was the wheel of Apollo I. Improved drift stability alone does not imply greater life.

"Therefore, the issue of attaching greater weight to differential watt meter reading, run-up and run-down time measurement, frequency profiles, dynamometer records, mass unbalance stability, or reduced voltage start capability, remains unsettled."

a. *Primary Guidance and Navigation System* - The IMU for the primary guidance and navigation system of the CM and LM consists of a three-gimbal platform mounting three SDF rate-integrating gyroscopes and three SDF accelerometers. The gyros establish a space reference to keep the stable element non-rotating such that accelerations are measured in a known inertial coordinate frame. Resolvers on the platform gimbal axes measure the orientation of the spacecraft relative to the stable element. Both the gyros and accelerometers are temperature controlled to +0.3°C utilizing a single temperature controller. One of the principal design problems was to achieve reliable temperature control of all inertial components under wide variations to the environment.

The platform gyros were chosen primarily to meet a 100,000-hour MTBF goal. A 1967 evaluation of production gyros indicated the MTBF goal had been met with 70 percent confidence (net 55). There have been no inflight failures, to date. Table 7 summarizes the Apollo gyroscope reliability experience through May 1971. Based on inflight measurement of gyroscope random drift, 9 out of 12 gyros in Apollo 7, 8 and 9 achieved better than 0.015 degree/hour performance.

The gyroscopes are calibrated during prelaunch activity and may also be recalibrated by the astronauts in flight against star sighting. During the Apollo 6 mission, a divergence was observed between

the attitude information supplied by the primary system and the backup strapdown system. This was first attributed to the primary system, but real-time review of prelaunch data on the backup system indicated that the measured drift would account for the divergence.

Table 7 Apollo Gyroscope Reliability Experience May 1971 (Block II-071 Configuration Gyros)

Number of Gyros	249
Bearing Failure	11
Total Gyro Operating Hours	370,800
45 Gyros Flown on Apollo Missions	
Preflight Wheel Hours	68,000
Flight Wheel Hours	4,400

2. Nimbus 3 Experience

Excessive drift of the Sperry gyroscope (gas bearing gyro) occurred during the Nimbus mission. The experimental test failures were due to shaft contamination, high gimbal torque due to relaxed ligaments, and motor failure due to scored thrust pads.

The gas bearing clearance of three to seven microinches becomes contaminated with outgassing polymers which tend to bind it sufficiently to prevent the motor from starting after being stopped for a period. The gimbal suspending ligaments tend to relax with time producing increased, and variable torque which determines the drift rate. The gyro drift error was excessively high; 20 degrees per hour, at the beginning of the mission.

3. Nimbus and Mariner

Two instances of Nimbus gyroscope bearing (ball bearing) friction (one complete gyro loss) and one instance of non-repeatable torques on Mariner. Kearfott Alpha type gyro was involved.

4. Biosatellite

In the Nortronics gyroscope (ball bearing) there was one instance of broken flex leads, two instances of null shift due to inadequate temperature compensation, three instances of contaminated flotation fluid, and one instance of fluid leakage.

5. Lunar Orbiter

In the Sperry ball bearing gyroscope during developemntal tests there were two instances of fluid contamination due to a bubble and to soldering flux causing gimbal hangup, two instances of broken pivots, and 12 instances of mass unbalance cured by coil form redesign.

6. Applications not Known

Gyroscope noise level gradually increased from launch until at 18 months (13,140 hours) it had increased by a factor of two. The cause is normal wear.

7. Titan Life Test

Six gyroscopes are being tested for life data. The purpose of the tests are to obtain life data concurrently with using; that is, items already on the shelf.

These gyros are single degree of freedom floated gyroscopes of the Kearfott Singer 2536 type.

S/N	Wheel Hours	Storage Time Hours
0000018	615.6	102
0000043	6634.4	100
0000052	204.5	99
0000053	6230	101
0000054	207.3	101
0000061	175.2	101

Serial No. 0000043 failed to meet specification for transfer function and wheel run-down time after 5029.5 hours. The transfer function subsequently shifted back in specification.

Serial No. 0000052 signal general null was outside specification limits. This has little significance because the value was on the edge of the tolerance band anyway and the total shift was not excessive.

Serial No. 0000054 was out of rate resolution (4.68 instead of 1.0 degree per hour) on Octover 7, 1971; however, on April 10, 1972 the rate resolution was within specification.

8. IDEP-ALERT

a. No. MSFC 69-05 Summary - Failure to start (lubrication breakdown). Of nine rate gyros taken from an eleven-month storage period, one wheel failed to start. Investigation revealed spin bearing lubrication distruction from heat and contamination. The cause was found to be improper heat sinking at the time the gimbal was originally sealed (welded). This idep alert is reproduced in Section I of this chapter.

9. Experience with Sterilization

Some investigation into some of the design aspects of gyroscopes is a result of sterilization studies on gyroscopes. Problems, such as corrosive properties of damping fluids, epoxy strength, absorption rates of materials, creep, and magnetic stability were emphasized.

In the aforementioned, the gyroscope motor, after 10 to 100 hours, failed to restart because of moisture in the gas bearing. Moisture in the gimbal parts evaporated into the gimbal atmosphere and was pumped into the gas bearing where it condensed. The failure mechanism allowed the rotor to "wring" the shaft. When the motor slowed down, the gas clearance approached zero. Motor was restarted when sufficient time was allowed for the water to evaporate again into the gimbal atmosphere. The corrective action consisted of drying the gas with a molecular sieve-type drying agent, along with the piece part drying prior to assembly.

Moisture lock-up was also prevented by using a bearing pattern that provides a flow through condition that carries the moisture away faster than it can condense. This is also done with an asymmetrical (different length) joined pattern.

The presence of organic vapor and particle contamination causes lockup. Sources of contamination can be detected by infrared analysis, ultraviolet-fluorescence analysis, mass spectrometry analysis, gas chromatograph analysis and thermal analysis test.

Contaminates are associated with bonding and encapsulating materials. The manufacturing process of vacuum bake can prevent the contamination.

10. <u>Martin Marietta Experience with Singer Kearfott Type 2536 (Martin PD96S008)</u>

In the Titan program, a Singer Kearfott 2536 rate integrating gyroscope is purchased as a component for installation in an electronic black box. The general performance of this gyroscope is at the lower end (less accurate) of the navigation class gyroscope. Failure reports covering some 600 of this type gyroscope has been reviewed. The general classification of the failures are:

1) Cable 15 events (cable design fix was instituted)

2) Administration - 44 events (miss test, known abuse, specification unclear, calendar life exceeded; calendar life revised during time).

3) Minor Specification - 49 events (slight out-of-tolerance and minor workmanship)

4) Excessive Drift - 9 events.

5) Solder Ball in Float Area-1 event

The following three events are life related:

1) Contaminated Jewel Bearing - 2 events (a new cleaning process was adopted)

2) Run-Down Time of Wheel was Excessive - 1 event.

3) Spin Motor was Erratic - 1 event; 51 months elapsed from date of manufacture.

11. <u>Control Moment Gyro for Skylab</u>

The control moment gyro (CMG) is used in the attitude pointing control system of the Skylab space station. It is primarily an actuator that produces control torques by use of the momentum generated in a large rotating mass. This spinning rotor is supported in a two-gimbal system. Early in the Skylab program, a design goal of 10,000 hours for the operating life of the CMG rotor spin bearings was established.

The CMG rotor spin bearings were designed to operate for 10,000 hours. The rotor weighs 63.5 kg (140 lb), runs at 821 rad/s (7850 rpm), and develops an angular momentum of 2713 kg m^2 s (2000 ft-lb-s). It was designed to survive the vehicle launch environment and possibly wide temperature extremes.

Each end of the rotor shaft is supported on a set of ball bearings. These bearings are lubricated with Kendall KG-80 oil supplied from the lubricating nut system, which is active only when the wheel is rotating. The complete bearing assembly is mounted in the bearing cartridge, which is a means of interfacing the bearing with the gimbal. The cartridge houses electrical sensors and heaters.

The lubricating system nut contains 10 g of KG-80 oil. The centrifugal force from the wheel rotation generates a pressure upon this volume of oil that forces the oil through a metered port onto the lip of the nut. Centrifugal force carries the oil from the nut lip to the step in the bearing retainer. From this step in the retainer, three holes carry the oil to the line contact of balls and races.

The interface of the lubricating nut to the ball-bearing assembly is made by screwing the nut onto the end of the wheel shift. The nut is locked in place with a retaining ring.

The metering element in the nut is a millipore restrictor 4.5 X 10^{-7}m (0.45 m) pore size. The restrictors are selected and installed to give a flow of 0.085 ± 0.035 mg/hr at 32°C. A design criterion was that no more than one-half of the total makeup lubricant would be expended at the end of 10,000 hr. This has been met on tests thus far.

Six bearing assemblies have been committed to a life test program--some with elevated temperature, subjected to vibration and normal laboratory environments. All gyros exceeded a life of 10,000 hours. Specifically, a unit at test at MSFC has operated over 29,000 hours without a revision in the oil supply. Another unit operated at elevated temperature (71°C) has operated over 27,000 hours with some "noise" in the bearing.

This program had two lots of bearings, one satisfactory, another was bad. It failed after several hundred hours because of "wetability". Test for "wetability" for this type of bearing appears to be required.

Generally, for large CMGs, this type of bearing appears to be satisfactory for the next 4 to 5 years. Careful process control on this proven design is required.

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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

ALERT

(Reporting Parts and Materials Problems)

1. PART/MATERIAL TYPE: Rate Gyro	2. ALERT NO.: MSFC-69-5	DATE: 4 June 1969 (DAY / MO. / YEAR)
3. MANUFACTURER AND ADDRESS: Nortronics, Division of Northrop, 100 Morse Street, Norwood, Massachusetts 02062	4. INDUSTRY/GOVT. SPEC. NO.: 50M35021	
5. REFERENCE: UCR #KSC 403058 (11-7-68) Report by J. R. Carter dated 12-31-68	6. MANUFACTURER'S PART NO.: GR-H4T	7. PART RATING OR SIZE: 0 ± 20 degrees/sec.
8. MANUFACTURER'S LOT OR DATE CODE:	9. MANUFACTURER'S SERIAL NO.: Package S/N 117 Rate Gyro S/N D642	10. CONTRACT AND/OR PURCHASE ORDER NO.: NAS8-13008

Rate Gyro

11. SPECIAL REQUIREMENTS OR ENVIRONMENT (Requirements placed on or extreme environment to which part/material was exposed.)

Temperature limits - -40°C to +74°C

12. LOCATION AND FUNCTION (Program, System, Subsystem and Component that utilized part/material and its intended function.)

Saturn AS-503, Control System, Control-EDS Rate Gyros

13. PROBLEM SITUATION AND CAUSE (State facts of problem and cause – failure mode and mechanism.)

The problem occurred on the AS-503 spare package, which had been in storage for eleven months. Three months prior to the launch date, the spare package was tested in the G&C Laboratory at the launch site. Of the nine rate gyros in the package, one wheel failed to start. Subsequent investigation showed the cause to be spin bearing lubricant destruction from heat and contamination. The cause was found to be improper heat sinking at the time the gimbal was originally sealed (welded). The operator had failed to follow manufacturing procedures while attempting to repair a leaky gimbal assembly, thus causing internal burning of epoxy and lubricant.

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14. ACTIONS TAKEN

The manufacturer, Nortronics, performed the failure analysis and attempted to reconstruct the events leading to the failure. Due to the time which had elapsed--two years--it was difficult to assess the severity of the problem, but it was concluded that the anomaly could have occurred more than once. Nortronics cooperated fully to tighten adherence to manufacturing procedures and they also performed adequate tests to show that if correct procedures were used there were no dangerous temperature rises.

MSFC has devised a wheel run up/run down test to detect any defective gyros. This test obviously shows wheel conditions only at the time of the test. The failure mode is time dependent, thus requiring the wheel tests to be performed shortly (4 months) before the intended launch. MSFC wheel test requirements are: (Continued on page 2)

22. ALERT NO. MSFC-69-5

15. RECOMMENDATIONS FOR FURTHER ACTION (Suggestions to prevent recurrence.)

Suggest that a test similar to MSFC's be performed, no earlier than 4 months prior to launch, on all units used in critical applications.

23. IDEP NO. 358.70.05.50-11-01

16. NAME OF INFORMATION SOURCE (Organization/Individual): MSFC/S&E-ASTR-CE, J. R. Carter; E. H. Fikes	TELEPHONE NO. AND AREA CODE: 205/453-5753; 205/453-0795	17. DATE MANUFACTURER NOTIFIED: 4 June 1969 (DAY / MO. / YEAR)
18. NAME AND AFFILIATION OF ALERT COORDINATOR: Elizabeth G. Manning, George C. Marshall Space Flight Center — TELEPHONE NO.: 205/453-3570	19. DATE SIGNED: 4 June 69 (DAY / MO. / YR.)	20. SIGNATURE OF ALERT COORDINATOR: Elizabeth G. Manning

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14. ACTIONS TAKEN (Cont'd)

Maximum time for gyro wheel run-up shall be 15 seconds after power application, up-to-speed indications shall be present and remain on until power removal.

Measured time for the wheel speed to decay from 100 revolutions per second (RPS) to 50 revolutions per second (RPS) must be greater than 4.0 seconds.

J. REFERENCES

1. J. T. McCabe and T. T. Chu: *Grease Lubricated Spiral Grown Gyro Bearing*. Franklin Institute Research Laboratories, Report F-C2028-1. October 1969.

2. R. A. Bovll: *Inertial Gyroscope System Applications Considerations*. MIT Instrumentation Laboratory Report R-645.

3. "Inertial Navigation - System and Components". AGARD, May 1968.

4. J. M. Bremer: *The Design and Build of a Gas Bearing Gyroscope Possessing High-G and Sterilization Capability and Utilizing a Low-Power Gas Bearing Spin Motor and High Frequency Pump*. Honeywell Aerospace Division. Report 20660-FRI.

5. W. Wrigley, W. M. Hollister: *Gryoscope Theory, Design, and Instrumentation*. MIT Press.

6. E. H. Fike: *Control Moment Gyro for Skylab*

COMPRESSORS AND PUMPS

by P. J. Powell

V. COMPRESSORS AND PUMPS

A. INTRODUCTION

The scope of this long-life study relates to pumps, compressors, and fans used in spacecraft thermal control systems, life support systems, environmental control systems, and ventilation systems. The components are grouped together and discussed according to their function. The pump failure modes, designs, and applications are discussed, followed by similar comments on compressors and fans. Compressors are classified as fans in several space programs (such as Skylab). Compressors, as considered in this study, are a gas flow producing machine which operates on the same basic principles as fans.

The long-life experience generated on these components has been limited to the manned spacecraft programs. The approaches to satisfying long-life requirements include: employing component redundancy, in-flight replacement of malfunctioning units, and developing long-life components.

Detailed comments concerning motors, bearings, and lubricants are covered in Chapter II - Electric Motors and Chapter IV - Gyroscopes and Bearings; and to avoid duplication are not repeated in this chapter.

To assist in understanding and accepting the prevalent life-limiting problems associated with these components, a brief description of the hardware and their functions are presented in Section F.

B. GUIDELINES FOR LONG-LIFE ASSURANCE

The survey of manufacturers and users revealed that continuous operation of compressors, fans, or pumps for 14,000 to 23,000 hours is within the present state-of-the-art. The life-limiting failure modes for these components are listed below in the order of problem magnitude:

1) Bearings and lubrication problems;

2) Seal leakage;

3) External housing leakage;

4) Structural failure of internal moving parts (fan blades, impellers, etc).

1. Design Guidelines

1) The optimum thermal control system design for long-life is a passive system. However, manned space missions require active systems which include such components as compressors, fans and pumps. Their applications are: molecular sieves, ventilation systems, coolant recirculatory systems, life support systems, waste management, and water systems.

2) All agencies surveyed considered bearings to be major life-limiting problem for compressors, fans and pumps. The operating life of compressors, fans, and pumps can be extended by accomplishing the following:

   a) Fabricating bearing material free from inclusions;

   b) Improving geometry (waveness) of the raceway;

   c) Improving surface finish, and;

   d) Maintaining lubricant thickness.

3) System designs which are adaptable to replacement of components (plug-in or snap-in type) are desirable and should be investigated for long life applications.

4) Require a vacuum melt or remelt process for metallic bearings. Metallic bearings must be free of inclusions or stringers for long-life applications.

5) Minimize the air gap in magnetic couplings to improve efficiency and promote long-life.

6) Use inert fluids for coolant system applications because of handling ease, they are dielectrics and they are good lubricants for system components.

7) Use wet motors (canned pump and motor) or magnetic couplings to avoid shaft seals which are a source of leakage, contamination and unpredictable drag.

8) Solve housing leakage in fluid systems by:

   a) Impregnation of castings with sealant substance;

   b) Using vacuum melt material to eliminate stringers or inclusions, and;

   c) Weld external pump housing joints.

2. Process Control Guidelines

1) The procurement specification pertaining to compressor, fan, and pump bearings (high speed, long-life, and heavily loaded) when used in failure critical applications, should contain a proviso for 100% dye penetrant inspection of balls prior to installation in bearings.

2) Use castings that are impregnated with a sealant or housing material that have been through a vacuum melt process to reduce inclusions or stringers.

3. Test Guidelines

1) Conduct 100 hour run-in tests prior to flight to eliminate components with latent manufacturing defects.

2) Conduct life endurance tests under operational conditions. For long-life applications, (data are not available for these units), the life parameters must be established.

3) Perform special bearing inspections to control quantity and contamination of lubricant. This action tends to narrow the wide variation of the applied lubricant.

4. Application Guidelines

1) The Gemini, Apollo and Skylab Program designs indicate that journal bearings have been used exclusively for pumps and that ball bearings have been used exclusively for fans and compressors.

2) The present approach does not develop long-life components, but selects components that have been proven on previous programs. Examples from the Skylab program, approximately eight months flight time, include:

   a) Coolant pumps are not replaceable; therefore the pumps are used in standby redundant configurations. Redundant coolant loops are provided.

   b) Redundancy and spare fans and compressors are employed.

5. Special Considerations

1) It is recommended that research into new state-of-the-art advances for long-life assurance include:

   a) The long term electrochemical effects of materials and fluids;

   b) The possible use of ceramic or air bearings instead of carbon-graphite for bearings;

   c) Methods to reduce wear particle generation;

   d) Material compatibility selection characteristics, and;

   e) Continuing effort for bearing development.

2) In spite of the adverse affects, current program decisions (such as Skylab) have been to power down systems during storage periods (approximately 30 to 60 days). Component starting and stopping transients will result in temperature excursions, distortions, stresses and cause part wearout.

3) A cost tradeoff study is needed to evaluate whether a concerted effort needs to be performed to extend the life of compressors, fans, and pumps or whether redundancy, maintenance, or restoration can be used on these components for extended missions.

C. LIFE LIMITING PROBLEMS AND SOLUTIONS

1. Failure Mode Analysis (FMA)

The failure modes that have been experienced by pumps, compressors, and fans are presented in the FMA (Table 1). The failure modes listed in the order of their most probable occurrence are:

1) Bearing Failure;

2) Seal Leakage;

3) Housing Leakage, and;

4) Structural Failure of Moving Parts.

All personnel interviewed generally agreed that the mechanical component failures would greatly exceed the electronic (solid-state) component failures.

A discussion of each of these failure modes is covered in the following paragraphs. Suggested solutions for these failure modes are discussed in the design section.

a. Bearings - The survey of manufacturers and users has revealed that bearings can be considered as one of the major problem areas that can prevent long-life thermal pump, compressor, or fan operation. The primary function of bearings are to maintain shaft or rotor alignment under radial and transverse loads.

The contributing factors that limit bearing life are:

1) Inadequate lubrication;

2) High speeds resulting in wear (boundary lubrication permitting metal to metal contact), and;

3) Incompatibility of bearing material and coolant fluid.

Table 1. Failure Mechanism Analysis - Compressors, Fans, and Pumps

PART AND FUNCTION	FAILURE MODE	REL. RANK	FAILURE MECHANISM	DETECTION METHOD	HOW TO ELIMINATE/ MINIMIZE FAILURE MODE
Bearings - Maintain shaft or rotor alignment under load. (See Chapter II for complete Failure Mechanism Analysis on both rolling contact and sliding surface bearings.)	Loss of power due to high starting or running torque	1	Dry film lubrication deterioration.	Slow starts and high motor currents.	1) Select stable dry lubricant. 2) Change to wet lubricant. 3) Reduce rpm for pump, compressor, or fan. 4) Utilize ball bearings, or
			Loss of wet lubrication.		1) Use grease lubricant instead of oil. 2) Use lubricant with low evaporation rate. 3) Change ball retainer to land type oil impregnated.
			Contamination.	Visual examination.	1) Assemble on laminar flow work bench. 2) Prevent foreign particles from entering component.
			Vibration	Erratic torque requirements.	1) Operate units during periods of high vibration (if possible). 2) Use sleeve bearings. 3) Employ vibration isolators.
Seals, static and dynamic - contain flow medium	Leakage	2	Wear or incompatibility of fluids.	Visual observation or inefficiency of unit	1) Dielectric coolant fluids. 2) Magnetic coupling. 3) Use redundant static seals. 4) Positive seal housing with welded construction.
Housing - support moving parts and contain flow medium.	External leakage	3	Housing porosity, stress corrosion, poor workmanship, cracked housings.	Visual observation, proof pressure tests for housings used in fluid systems.	1) Inpregnate castings with sealant. 2) Use bar stock that has vacuum melt treatment. 3) Weld housing parts. 4) Increase housing design margins.

Table 1. Failure Mechanism Analysis - Compressors, Fans, and Pumps (concl)

PART AND FUNCTION	FAILURE MODE	REL. RANK	FAILURE MECHANISM	DETECTION METHODS	HOW TO ELIMINATE/ MINIMIZE FAILURE MODE
Failure of rotating or sliding part (impellers, fan blades, etc.) pass flow medium through	Structural failure	4	Fatigue, overstressed part.	Visual observation.	1) Use large safety margins. 2) Eliminate stress concentration points

Problem - The limited knowledge on long-life bearings can be traced directly to "There has been no requirement." To date, maximum operational life requirements on aerospace hardware, such as pumps, compressors, and fans, have been 1000 to 2500 hours. The most severe continuous operation requirements stipulated has been the 14 days in a 5-psi environmental system for the Apollo space capsule. The eight-month requirement for the Skylab program, includes five months of active operation.

In the hard vacuum of space it was generally conceded that conventional lubricants are not satisfactory for long duration missions. Lubrication depletion and deterioration was by far the most common cause of any bearing failure. Component long-life bearings and lubrication problems are described in Chapter II. Pump designers have approached the problem by using the coolant fluids as the bearing lubricant.

The space environments require specialized fluids that rule out the more conventional antifriction metal bearings. These specialized fluids have either very poor lubrication qualities or are extremely corrosive to the standard metal bearing. High-speed operation of the pump impeller shafts accelerate bearing heating, requiring either a large lubrication reservoir or other suitable heat sinks.

b. Seals - Two types of seals are used in thermal control pumps; static and dynamic. The dynamic seals are being eliminated by the use of magnetic couplings and the use of dielectric coolant fluids.

Problem - This survey revealed that long-life static seals could be a great problem. The factors that limit the life of seals are:

1) Compatibility of seals and fluids;

2) Fluid expansion and contraction, and;

3) Contamination.

c. Housing - The housing can be made by machining, casting, or forging. Although castings can be used for various shaped housing, their material properties are not as good as those made from bar stock or forgings.

Cracked housings occur at fillets, flanges, and webs and are caused by inadequate design margins as a result of meeting minimum weight requirements.

Problem - The factors that contribute to housing failure are:

1) Casting or housing leakage;

2) Stress corrosion cracking of housing; and

3) Cracked housings.

d. Structural Failure of Moving Parts - The moving parts include impellers and fan blades. Flutter of fan blade can cause repeated flexure and fatigue.

Problem - The long-life factors that contribute to structural failure are:

1) Fatigue, and;

2) Over-stress.

2. Design

a. Selection Criteria - Comparative design features (advantages and disadvantages) from pumps, compressors and fans are discussed in Table 2. The long-life assurance design factors for the basic parts which make up these components are further discussed in Table 3.

The main efforts of this study are directed toward increasing the life of the individual parts that make up the pumps, compressors, and fans. The design approaches for extending the life of these parts and eliminating the previously discussed failure modes are listed below. In addition, materials compatibility and fan noise suppression are also discussed.

Bearings - Within the last few years, thermal pump operational requirements have been continually increasing. New coolant fluids have been developed to satisfy these new requirements. Several new bearing materials have been developed to be compatible with the fluids. For example, carbon-graphite (ESSO) bearings that are lubricated by Freon-21 have been tested for 4400 hours and showed minimal wear or deterioration. Care must be exercised to prevent contamination of the bearing lubricant to prevent excessive wear.

Rulon A and Vespel SP21 bearings have been successfully developed to be used in fluids with high temperature (near the refrigerant breakdown point).

A recent Norton Company news release which covered the Federal-Mogul test program is encouraging and indicates that for the first time ceramics have the potential of outperforming a high performance steel bearing. However, the data was generated from four or five ball tests; and, they are in the development stages of their program with ceramics as anti-friction bearings.

For high speed fans, conceivably an air bearing may be a valuable bearing for applications above 30,000 rpm's. Detailed comments for air bearings are covered in Chapter IV - Gyroscopes and Bearings.

The Lubrication Handbook For Use In The Space Industry (Midwest Research Institute, Kansas City, Missouri, March 1972) is a suggested reference to provide a ready information source for many of the solid and liquid lubricants used in the space industry. Developments in fluoroether and polyether lubricants have shown these synthetic fluids to be satisfactory for the hard vacuum of space. These lubricants are highly tolerant to evaporation, migration, have no tendency to form deposits or washout, as well as being non-corrosive and capable of maintaining the required film thickness under load.

Testing of the perfluoro ethers at Battelle Laboratories[1] below 40°C would satisfy NASA requirements to protect vacuum-ultraviolet optical systems from contamination due to outgassing of solid materials.

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[1] D. B. Hamilton and J. S. Ogden: The Evaporation of Various Lubricant Fluids in Vacuum, ASLE paper, preprint number 72LC-6C-2, October 1972.

Table 2. Comparative Design Factors

Component	Advantages	Disadvantages
Piston Pumps	Efficient for small rates of discharge or capacity, and high pressure and suction lifts.	Not suited for the handling of very viscous fluids or dirty media, due to the tendancy to clog suction or discharge valves. Dynamic seals. Also pulsating flow and weight.
Rotary Pumps (vane, gear, and lobe types)	These pumps do not have valves (as the piston pump) and can handle thick viscous fluids. Are suitable for service under low to medium head.	Not particularily suited for handling grit or abrasive materials due to the close tolerances between the rotary element and the case. Limited use for non-lubricating fluids with low viscosity such as water.
Centrifugal Pumps	Simple, long life, compact, low cost, and can operate under a variety of conditions.	Less efficient than the rotary pumps. Low head per stage. Turbulent flow, (Ref 1)
Axial Fans	High efficiencies, in line flow, motor cooled by air-stream, compact total envelope.	Low pressures developed (0-8 in. of water).
Centrifugal Fans (Compressors)	Stable performance at low flow rates, built in 90° turn from inlet to outlet, highest pressure developed.	Medium pressures developed (0-25 in. of water).

Table 3. Design Factors for Long-Life Assurance Part/Component: Pumps, Compressors, and Fans

Design Factors	Remarks
Bearing Material	Flow media and lubricant are the most important considerations (see text for particular recommendations). Ceramic bearings may be required for long life. Metallic bearings must be free of inclusions or stringers.
Dynamic Seals	Avoid dynamic seals which wear and cause contamination by using: 1) Magnetic coupling system. 2) Submerged pump/rotor assembly. 3) Totally wet pump/motor.
Static Seals	Brazed or welded housing joints are preferred to captive types of seals for fluid systems.
Materials Compatibility	Inert fluids are recommended, such as Coolanol or Oronite for coolant systems.
Housing	Housing leakage in fluid systems can be solved by: 1) Impregnation of castings with sealant substance. 2) Using vacuum melt material to eliminate stringers or inclusions.
Noise Suppression	Methods to suppress noise include: 1) Mechanical isolation. 2) Sound suppressor/acoustical insulation material. 3) Non-metallic duct and connectors.
Journal Bearings	<u>Advantages</u>: simple, inexpensive, and can be used in small spaces. <u>Disadvantages</u>: higher coefficient of friction than ball bearings
Ball Bearings	<u>Advantages</u>: low friction, short length, can accept both radial and thrust loads. <u>Disadvantages</u>: diametrically large, costs more than journal bearings.
Roller Bearings	<u>Advantages</u>: higher load capacity than ball bearings, diametrically smaller than ball bearings. <u>Disadvantages</u>: longer than ball bearings, costs more than journal bearings.

The final solution will depend on the requirements of the customer. The state-of-the-art exists to develop the bearings and the matching of fluids; but because of expensive testing, the companies will not develop and test for long life until long lives are required.

Seals - The pump manufacturers are using elastomers that are compatible with the coolant fluids, such as:

1) Nitrile;

2) Butyl rubber;

3) Chloroprene (Neoprene);

4) Silicone rubber;

5) Fluorocarbon rubber (Viton), and;

6) Ethylene propylene.

There are, of course, no final solutions. As better fluids and pump designs are developed, the static seals of better design and improved fluid compatibility will keep pace. It is recommended that a continuing review of the state-of-the-art be implemented by Martin Marietta's Denver Division (Ref 2) for NASA.

Dynamic seals are being eliminated by the use of magnetic couplings that transmit ac power to an ac induction motor that is canned and runs completely submerged in the coolant fluid. The coolant fluid is dielectric. The approach has been incorporated into the Apollo Lunar Module (LM) coolant package.

Figure 1 shows the currently used motor and pump coupling arrangement. They are:

1) Totally wet motor - does not require a sleeve around the rotor; the coils are encapsulated and non-corrosive coolant fluids are used.

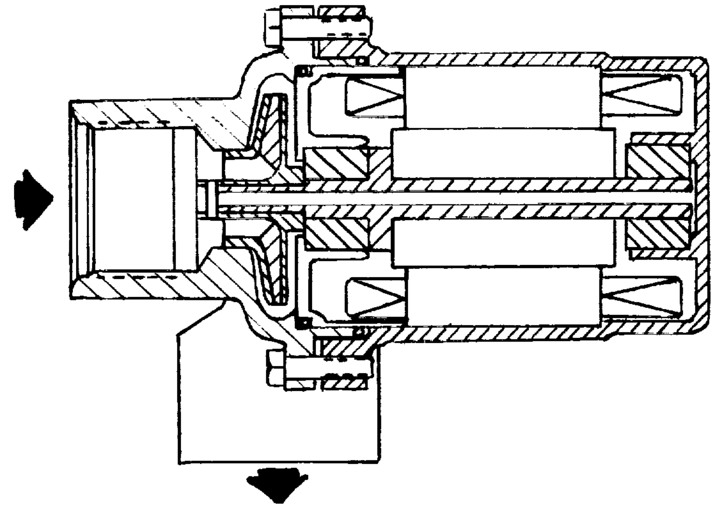
2) Submerged pump and rotor assembly - a sleeve is located around the rotor assembly and the coils can be hermetically sealed.

3) Magnetic coupling system - the advantages are:

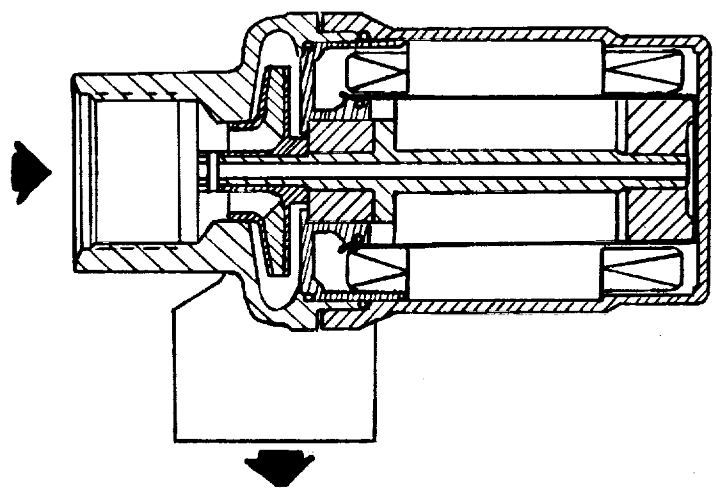
   a) No shaft seal;

   b) No viscous drag of the motor rotor (better efficiency), and;

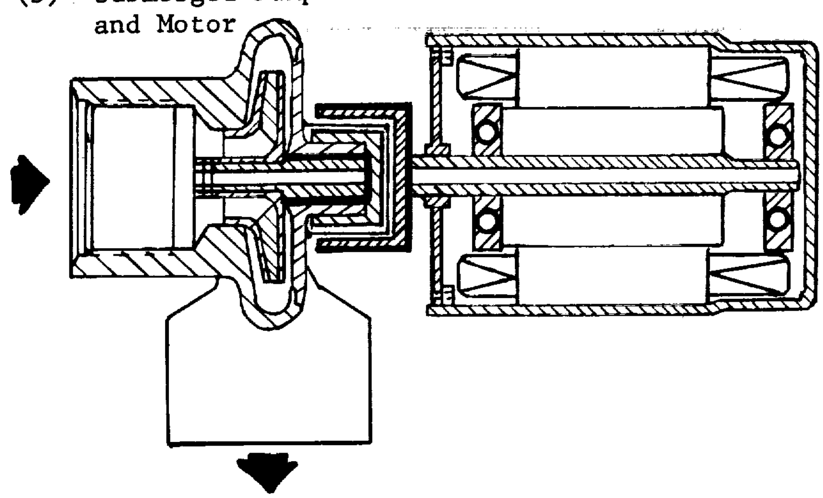
   c) Dry motor.



(a) Wet Motor



(b) Submerged Pump
and Motor



(c) Magnetic Coupling

Figure 1 Pump Coupling Arrangements

High-Strength Materials - Inherent in the development of high-performance turbomachinery is the need for high-strength materials (Ref 3). This involves the selection or development of suitable materials and the development of acceptable production techniques.

The general survey of manufacturers and users revealed that high-strength materials represent a continuing area of research; high-strength materials are not considered a long-life problem that is beyond the state-of-the-art, but rather a research and test problem.

Effort is being expended in the development and selection of materials and coatings for stress corrosion protection, the development of joining of dissimilar metals, and the investigation of advanced alloys, such as columbium, beryllium, molybdenum, etc. These are areas that require surveillance and continuing evaluation of advances in the state-of-the-art.

System design and fluid compatibility with the design material are of primary importance if component long-lives are to be achieved. Important considerations in the design of long-life are:

1) Development of long-life bearing;

2) Development of adequate seals;

3) Evaluation and selection of high-strength materials, and;

4) Better flow patterns and smaller energy losses.

Data obtained from the "Parts, Materials, and Processes Experience Summary" (Ref 4) provides users with the accumulated experience resulting from ALERT reports issued by NASA and the Government-Industry Data Exchange Program (GIDEP). The reporting period covers from 1964 through March 1971. Only one problem was reported on compressors, fans, and pumps. The problem was that of cracks appearing at the same point and at the same angle on the fan blade. Problem analysis revealed that flutter induced by the leading edge of the blade causing a constant flexing at the rivet point creating a crack which elongates with centrifugal force and constant flexing. The problem was corrected by subsequent redesign and structural reinforcement of the fan blades.

Materials Compatibility - The typical coolant pumps used in aerospace systems have shafts that are very small (less than 1/4 inch diameter) and the bearings are not hydrodynamic. The bearings operate in the boundary regime and fluids are fed to the bearings for lubrication and cooling. Suitable bearing materials must be selected and are often carbon-graphite.

Bearing material selections are the main design problems for pumps, compressors, and fans. Typical utilization of bearings are shown below:

1) For water glycol systems, carbon graphite pump bearings are rated good;

2) For Coolanol systems, carbon graphite pump bearings are rated fair;

3) For other fluids, glass filled Teflon pump bearings can be used for 10,000 to 20,000 rpm applications, and;

4) Ventilation fans studied have employed ball bearings.

The survey of manufacturers and users indicated they are not completely satisfied with water systems (which are usually adopted to provide an emergency drinking water source). Although ionized and purified, water remains a very reactive solution involving corrosion and the dissemination of sediment in pump clearances after draining. Continuous operation presents little problem, it is the dwell periods that permit chemical reaction to proceed and materially affect the pump and fluid system components. One particular contaminant isolated was Reevesite.

Metal wear products form the nucleus of chemical reactions and the carbon graphite bearings also react with the coolant. The inhibitors such as Maridyn and Raccol have improved this situation, but cause increased wear. Chelates was another consideration where the abrasive nature was unknown along with the quantities needed for metallic ion control. The ΔP and Δt's that result from the systems restrictions, particularly the relief valves, accelerate chemical reactions in the water systems. The problems identified with the water systems are minimized by the use of the centrifugal pumps.

Hence, it is recommended that water systems not be used, whenever possible, particularly with pumps involving close fitting parts (vane fit are generally around 0.0002 in.). However, several systems onboard a spacecraft (potable water, cabin thermal systems,

and etc) require the use of a water pumping system. Inert fluids, such as Coolanol or Oronite should be used whenever possible. Coolanol fluids are particularly easy to handle since it is a dielectric and is a good lubricant. A wet motor (uncanned) can be used with this medium.

For these thermal control systems it was concluded that the centrifugal type pump was to be preferred, but cannot always be employed because its efficiency is satisfactory for only a certain specific speed range. See Figure 2 (from Ref 5) Efficiency versus Specific Speed.

The specific speed is obtained by its relationship to rpm, flow rate, and head. The classifications shown in the figure are centrifugal, mixed flow, and axial flow pumps. These are classified according to the flow path through the pump. By multistaging centrifugal pumps with the overall ΔP split between pumps, the centrifugal pumps can be placed in a more satisfactory efficiency position.

For comparison with the pumps, the Efficiency versus Specific Speed for various types of fans is presented in Figure 3 (from Ref 5). The fan efficiencies are lower and the specific speeds are higher than the pumps studied.

Noise Supression - The technique used for noise suppression is to prevent the transfer of vibrational energy to large surfaces which dissimate this energy into the air in the form of acoustical radiation. The amount of sound emitted by moving parts are mainly due to the vibration from the supports and structure. The amounts emitted to the air are negligible. Materials of different specific acoustical resistance are used between moving parts and supports to reduce the transfer of noise, and the control method is dependent upon the frequency of emission. Methods used for fan noise suppression include:

1) Mechanical isolation;

2) Sound suppressors;

3) Non-metallic ducts and connectors, and;

4) Acoustic insulation materials.

Incorporate 1) through 4) to reduce the noise level to 55 decibels in the crew areas for manned missions.

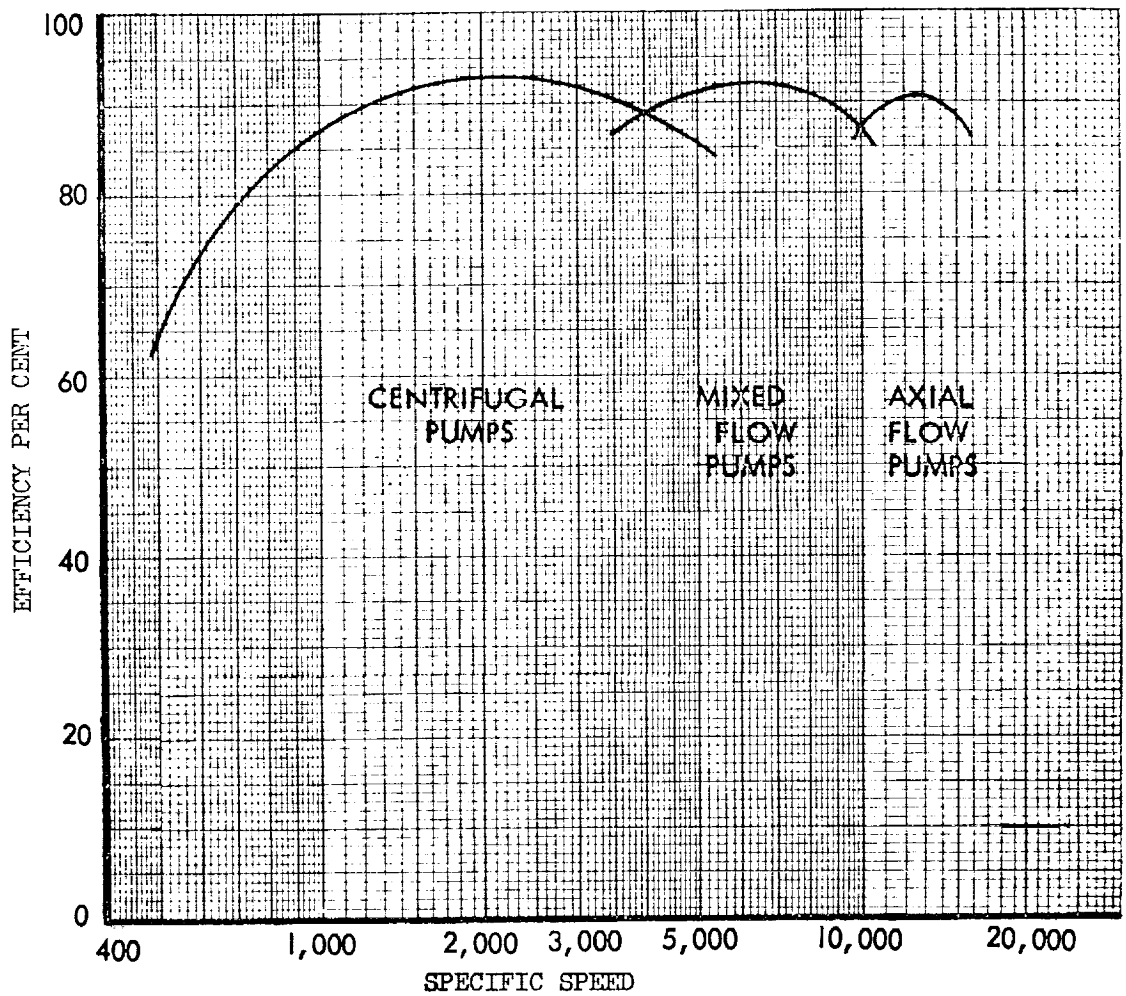


Figure 2. Pump Efficiency

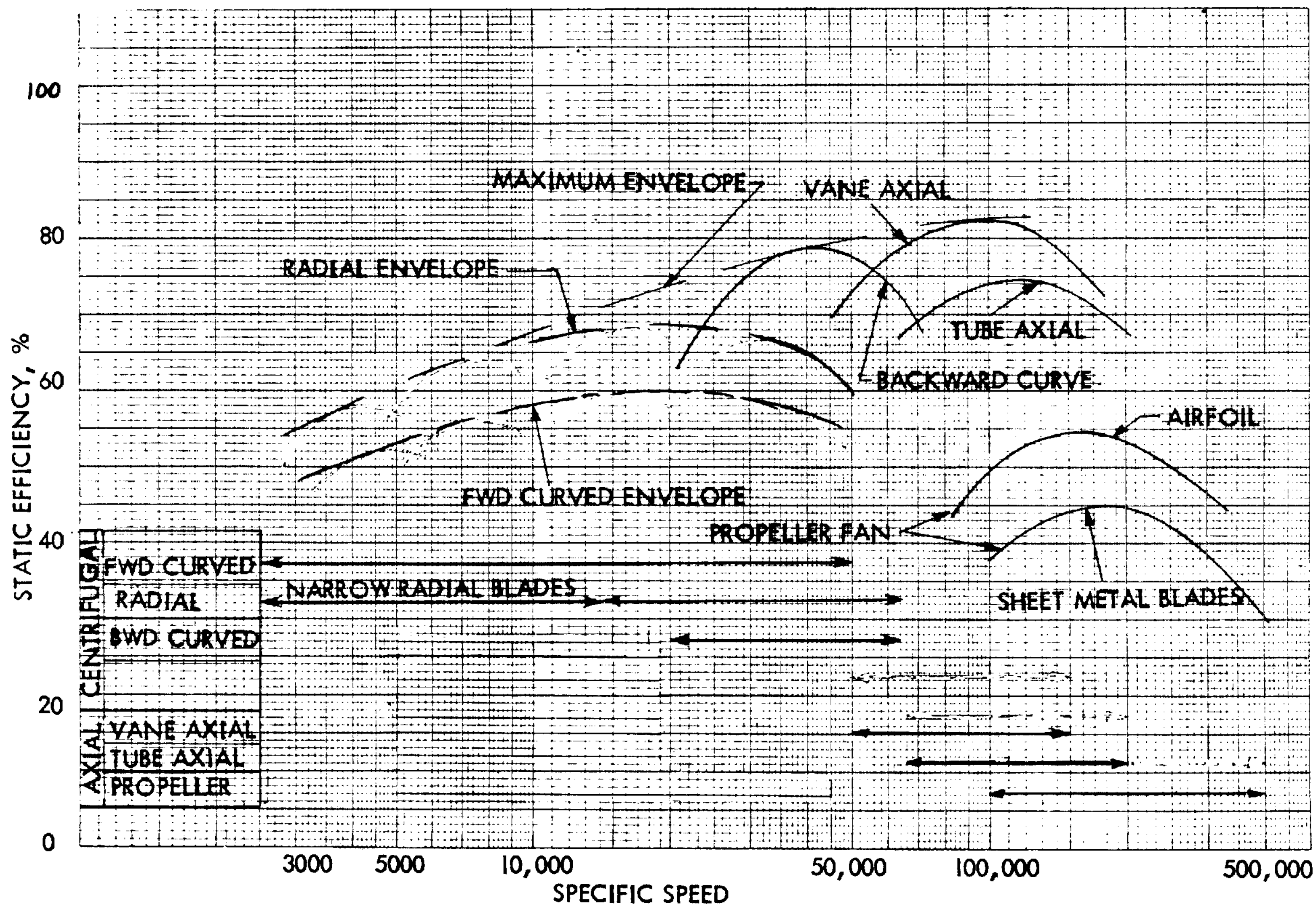


Figure 3. Fan Efficiency

b. Survey Results - Two surveys were conducted. One survey involved general questions concerning the long life assurance of compressors, fans, and pumps; the consensus of the answers is presented in Table 4. The second survey determined why specific selected compressors, fans and pumps obtained long lives and high reliabilities. The results of this survey are summarized in Table 5. Response from both surveys are incorporated in the discussions throughout this study.

The survey indicates that none of the compressors, fans, or pumps have been developed for long lives such as five years or longer. The components have been developed for manned flights. The Apollo flights have been up to 14 days duration, the Skylab mission is approximately 8 months duration of which roughly 5 months are manned. As a result there has been no long-life space requirements for these type of components until now.

Both the users and manufacturers surveyed indicated that the bearing/lubrication combination were the life limiting factors for compressors, fans, and pumps. Contributors to the problem were:

1) Grease depletion due to time and temperature resulting in excessive wear or galling of bearing interfaces;

2) Grease blowout through the bearing shield;

3) Eccentric loading or misalignment of shaft, and;

4) Wear combined with contamination and loading to fatigue bearing.

An observation from this study is that fans and motors generally use ball bearings and pumps used plain or journal bearings. The reason for these applications are:

1) More viscous drag is present on ball bearings if they are submerged in fluid;

2) Journal bearings are lubricated by the coolant fluid, and;

3) Journal bearings are smaller and would require smaller pump housings.

Table 4. Results of General Manufacturing/Agency Survey for Compressors, Fans and Pumps

QUESTIONS	
1. Do you manufacture (use) aerospace compressors, fans or pumps? Usage?	Gemini, Apollo, LEM, Skylab
2. What is the expected life of subject part?	Ventilation fans 20,000 to 25,000 hours. Compressors (rotary) 4,000 hours. Pumps, 14,000 hours.
3. What are the failure mechanisms (causes) of the failure modes?	Bearing problems due in part to vibration, temperature, and lubrication.
4. What failure modes would prevent a ten year service life? What failures have occurred?	1. Grease depletion from bearing. 2. Grease blow out of bearing shield. 3. Eccentric loading of bearings. There has not been any recorded flight failures.
5. What solutions do you suggest for the above failure modes that would either enhance the operational life and/or increase the probability of success?	1. Increase bearing size 2. Increase bearing preload 3. Lower shaft speed 4. Use a larger motor 5. Perform preventative maintenance.
6. To achieve long life, what design features are incorporated in your unit?	1. Should require high fatigue factor for bearings. 2. Maintain lubrication film.
7. How do you determine part life?	1. Life tests or endurance tests. 2. Determine margin above mission requirement.
8. Did you test for specific failure modes and mechanisms and were any special testing techniques used?	1. Lubrication tests 2. Fire hazard tests 3. Contamination tests
9. What process controls are necessary to ensure long life?	1. Sample bearings to verify proper lube quantity. 2. Inspect lube for contamination levels. 3. Component run-in. 4. Traceability of piece parts.

Table 5. Survey of Users and Manufacturers on Specific Parts

PART DESCRIPTION AND USING PROGRAM	USERS OPINION OF WHY PART WAS SUCCESSFUL	MANUFACTURER'S OPINION OF WHY PART WAS SUCCESSFUL	RECOMMENDED GUIDELINES IMPROVEMENTS, PROCUREMENT SPEC REQUIREMENTS, ETC.	RELATIVE IMPACT ON YIELD & COST OF INCORPORATING RECOMMENDATIONS
COOLANT VANE PUMP USED ON THE SKYLAB PROGRAM.	1. Water system problems reaction between metals and water if dried out. 2. Tried to keep pump wet and clean. 3. Close fitting parts are difficult to flush or dry after foreign mater is generated.	This unit has passed qualification testing, however water remains a very reactive solution involving corrosion and precipitation sediment in pump clearances after draining. Continuous operation presents little problem, it is the dwell or idle periods that permit chemical reaction to proceed and materially affect the pump and other water system components.	1. Recommend that water systems should not be used. (Use inert fluids such as Coolanol or Oronite). 2. Do not use close fitting components in water systems (such as a vane clearance of .0002"). 3. Require absolute filters upstream of pumps, that would eliminate elongated contamination substance to pass through the filter.	This pump has been used with ethylene glyool on the LM Program, so there would be no cost change for another application. The filter would not be a pump delta cost, but a delta system cost.
COOLANT PUMP USED IN THE LM	1. This is used in a ethylene glycol fluid, which is inert and aids in lubrication and cooling of the pump. 2. Uses a magnetic coupling which eliminates the need for a pump shaft seal. 3. Pumps are used redundantly. 4. Life margin of Approximately 14:1. 5. Approx. 125 hours of ground checkout prior to flight.	1. Agree with the users comments. 2. Vane type pumps were chosen because of efficiency. 3. These are direct motor driven pumps using canned motors and plain journal bearings.	1. They recommend that water systems should not be used and not with pumps involving close fitting parts. 2. Recommend journal bearings for long life. 3. Require a run-in of 100 Hrs. 4. Tests required by Users procurement specification included: a) Helium leakage of submerged portion of pump. b) Contamination tests c) Start and stop tests d) Vibration e) Circulated water as a test fluid with specific contamination levels.	The manufacturers are no longer interested in fixed price programs but would consider fixed payment for specific segments of work.

Table 5. Survey of Users and Manufacturers on Specific Parts (cont)

PART DESCRIPTION AND USING PROGRAM	USERS OPINION OF WHY PART WAS SUCCESSFUL	MANUFACTURER'S OPINION OF WHY PART WAS SUCCESSFUL	RECOMMENDED GUIDELINES IMPROVEMENTS, PROCUREMENT SPEC REQUIREMENTS, ETC.	RELATIVE IMPACT ON YIELD & COST OF INCORPORATING RECOMMENDATIONS
Waste Management System (Area). Compressor used on Skylab. Backward curved blade, 2 stage, impeller at each end, 25,000 rpm.	1. Application does not require a long operate time, but many start and stop cycles. 2. Used Kry-tox 288 lube on bearings.	1. Special application and this unit exhibits a good life for this class of hardware. 2. They consider this compressor an optimum design.	1. Recommend that instead of trying to improve this particular compressor, that spares should be made available.	Develop a new design if longer life is required.
Cabin fan, brushless dc motor driven, axial flow fan. This fan is used on the LM vehicle.	1. Brushless dc motor provides good efficiency. 2. Failure history of fan indicated that there had been no previous in-flight failures of this fan. 3. Extensive testing to check for fire hazard (jammed rotor test and voltage overload tests). 4. Grease evaluation tests.	1. Modified design of a proven unit. 2. There is a large life margin on this fan. 3. Application of hardware proven successful. 4. Materials testing, quality control activities and traceability of piece parts.	1. Examine bearings for possible improvement to extend for life. a) Increase bearing preload b) Provide for lubrication retention on bearings. c) Develop better material for bearings. 2. Require life tests.	<u>Non-Recurring</u> 2K for demonstrated life test (1 unit) <u>Recurring Costs</u> The R&D for long-life bearings is a continuous cost.

Table 5. Survey of Users and Manufacturers on Specific Parts (cont)

PART DESCRIPTION AND USING PROGRAM	USERS OPINION OF WHY PART WAS SUCCESSFUL	MANUFACTURER'S OPINION OF WHY PART WAS SUCCESSFUL	RECOMMENDED GUIDELINES IMPROVEMENTS, PROCUREMENT SPEC REQUIREMENTS, ETC.	RELATIVE IMPACT ON YIELD & COST OF INCORPORATING RECOMMENDATIONS
The coolant pump assembly consists of a fill valve, two poppet check valves, a coolant reservoir with a low level switch, two gear pumps, two electrical motors, fill port, inlet port, and outlet port all installed in a common housing. This pump assembly is used on the Skylab Program.	1. Agree with manufacturers comments. 2. Self contained accumulator at pump inlet. 3. Internal cooling of rotor. 4. Magnetic Coupling. 5. Carbon bearing in motor and journal bearing pump. 6. Extensive testing of hardware parts that comprise the pump package assembly.	1. The positive displacement gear pump is driven through a direct coupling. 2. Pumps are used redundantly in the coolant assembly. 3. High vibration levels are not a problem for the plain journal bearings. 4. Successfully used as a part of the total Gemini coolant package.	1. External case is welded after final assembly to preclude leakage. 2. Don't truncate life tests, that is, run these tests until the unit is out of limits or fails. 3. Develop an accelerated life test for pumps. 4. Develop long life bearings.	The margin life tests would increase the cost of the life test (during qual test) by a factor of 10.
The compressor consists of a single stage centrifugal impeller attached directly to an induction type electric motor inside an aluminum housing. This unit is used on the Skylab Program.	1. Used redundantly. 2. An aluminum alloy conduit is provided for the electrical leads across the flow path to the exterior of the compressor. 3. A hermetically sealed electrical receptacle is provided. 4. Although failure history of this unit is good--spares are provided. 5. The unit was subjected to extended ground checkout of roughly 14 days.	1. Compressor was operated at a modest speed, air flow kept temperature of component low, precise aligned rigid shaft, and two greased-packed ball bearings support the squirrel-cage induction motor rotor and the extended centrifugal aluminum impeller.	1. Recommend ground checkout of compressor (100 hours run-in) prior to flight. 2. Require life endurance test. 3. Develop long-life bearings.	<u>Non-Recurring Costs</u> 2K for demonstrated life test (1 unit). <u>Recurring Costs</u> The R&D for long-life bearings is a continuous cost.

Table 5. Survey of Users and Manufacturers on Specific Parts (concl)

PART DESCRIPTION AND USING PROGRAM	USERS OPINION OF WHY PART WAS SUCCESSFUL	MANUFACTURER'S OPINION OF WHY PART WAS SUCCESSFUL	RECOMMENDED GUIDELINES IMPROVEMENTS, PROCUREMENT SPEC REQUIREMENTS, ETC.	RELATIVE IMPACT ON YIELD & COST OF INCORPORATING RECOMMENDATIONS
Ventilation fan used throughout the cluster on the Skylab Program	1. Successful history on the Apollo Program. 2. Require positive pressure in the cluster at all times to retain lubricant on bearings. 3. Provide 100 hour ground check out prior to flight. 4. Provide spare fans for in-flight maintenance activities.	1. Agree with user's comments. 2. Large ball bearing, spring pre-load on bearing, and airflow provides cool running fan. 3. Had completed 10,000 hour lube tests for lube selection. 4. Similarity to existing successful hardware. 5. Exhaustive piece part testing (i.e., bearings and lube).	1. Require life endurance tests to determine fan life. 2. Don't expose the bearings to vacuum. The loss of bearing lube would increase bearing friction and cause wearout. 3. Provide provisions for fan replacement. This fan has undergone a design improvement program where the following changes have been incorporated: a) Increased bearing size, b) Increased bearing preload, c) altered fan housing to allow in-flight replacement.	<u>Non-Recurring</u> 2K for demonstrated life test (1 unit) Cost is required for each spare fan plus the storage location. <u>Recurring</u> Cost of handle for replacing fan is negligible.

Most life endurance testing was truncated because the mission requirements were exceeded or because of time and cost restraints. It is recommended that these type tests be run until the component fails (margin testing). This would provide positive parametric life data, but may require accelerated testing techniques.

c. Alternate Approaches - The alternate approaches of thermal control systems involve the selecting of the type to be used. Refrigerant, coolant, heat pipes, or passive thermal control systems, represents some of the choices available. Spacecraft applications include: refrigeration systems for food storage (Skylab Program); coolant loops have been used for flight crew suit cooling and electronic cooling; heat pipes uses are structure, electronics and Orbiting Astronomical Observatory (OAO) telescope thermal control; and passive thermal control techniques have been widely used and the methods involve paint, insulated wrap, and louver panels.

The types of fans or compressors used for a particular application depends on the magnitude of the resisting pressure against which the air is moved. Although no strict demarcation exists, and definitions have not been standardized, the following table indicates the range and type of air handling equipment capabilities (Ref 1).

Propeller Fans	0 - 8 inches of water
Centrifugal Fans	0 - 25 inches of water
Blowers	1 to 5 lb/in^2/stage
Centrifugal Compressors	30 to 150 lb/in^2

Additional Studies Required - From the survey of manufacturers and users, it was determined that the possible studies and test programs for research into state-of-the-art advances for long-life assurance include:

1) The long term electrochemical effects of materials and fluids;

2) The possible use of ceramics instead of carbon-graphite for bearings;

3) Methods to reduce wear particle generation;

4) Material compatibility selection characteristics, and;

5) Continuing effort for bearing development.

d. Hardware Life - From the survey of manufacturers and users the current life estimates for the following components are:

1) Pumps, 14,000 hours;

2) Fans, 20,000 to 23,000 hours, and;

3) Compressors, 4,000 to 6,000 hours.

For this analysis, a spacecraft thermal control pump has been defined as a fluid recirculating pump that is capable of operating from the spacecraft's electrical system, usually 28 to 32 Vdc (but also may include 110 Vac, 400 Hz, or 200 Vac, 400 Hz). The pumps, fans and compressors are generally characterized by their high impeller speed of 5000 to 30,000 rpm, small size, light weight, and small power requirements.

During this analysis, only two manufacturers were located who design, develop, manufacture, and test thermal pumps for spacecraft application; Garrett AiResearch Manufacturing Division and Hydro-Aire, a Division of Crane Corporation. Both manufacture a pump assembly package consisting of multiple pumps. Normally, the capacity of any one of the pumps is sufficient to satisfy the system requirements.

Both companies stated that the state-of-the-art exists to build thermal control pumps that would operate continuously for one year, and each was the opinion that three years operation is possible with development and substantial testing. It is possible that the present pump design could be modified to obtain a three-year life; testing is a big problem because of the time involved within endurance testing. It is thought that the compressor life may be much longer than the data indicates, however, in many cases the test for compressors, fans, and pumps has been truncated.

This survey of manufacturers and users revealed that the problems involved in developing high-performance long-life thermal control pumps are:

1) Development of long-life bearings;

2) Development adequate seals (loss of flow media could limit mission duration);

3) Development of high-strength materials;

4) Determination and use of compatible fluids, and;

5) Determination of flow patterns.

The problems involved in developing high-performance long-life fans and compressors are 1), 2), and 3) above.

Note that the operating environment is primarily the pacing consideration in the design of the thermal pump. Size, weight, and coolant fluid all force design approaches, but in the final analysis it is the operating environment that determines the final design.

A Planning Research Spacecraft Study (Ref 6), which include thermal control systems, listed no failures for compressors or pumps during launch or orbit activities. These statistics are based on 304 spacecraft used on 41 programs. The spacecraft data includes a sample of 48 units and 5100 total survival hours in orbit for compressors and pumps. However, these data are from short duration programs and problems develop when operating these components for long duration flights of ten years (87,600 hours). Specific life data and applications for various compressors, fans, and pumps are presented in the following paragraphs.

The suit compressors used in the Apollo Command Service Module (CSM) are basically the same compressors that are used in the molecular sieve on the Skylab program. The Apollo unit compressor design life was 1200 hours and the life data ranged from 2124 to 3125 hours. The mean operating time to failure was 2673 hours, based on six units tested. The Skylab molecular sieve compressor design life was 2000 hours and one unit has experienced 3743 accumulated operating test hours.

Two Gemini fans were operated for 20,879 and 22,195 hours each before failure. The design life for these fans was 2000 hours. The Skylab circulation fan is basically the post-landing ventilation fan used in the Apollo program. The design life for the Apollo fan is 1200 hours. The mean time between failure for two units tested were 4750 and 8317 hours. Two other fans were tested to 4738 and 21,888 hours and the tests were terminated without fan failure. The Apollo fan housing was modified for replacement ease and the fan was furnished to the Skylab program as Government Furnished Equipment.

The early Apollo glycol pumps used ball bearings, rather than the currently used journal bearings. The test program from which these data were extracted was conducted primarily to evaluate the longevity of the glycol pumps. Four pump assemblies were tested and the pump, impeller, housing bearings and check valves exceeded the 1200-hour minimum operating time requirement for

the Apollo mission in some cases by a factor of 10. The glycol pump motor life of 7000 hours was noted for the early design, but improved motor bearings have been tested for 10,000 hours (Ref 7).

McDonnell Douglas have operated these gear type pumps over 9000 hours during Skylab testing. The vane type water pumps that were used on LM have been subjected to 6000 hours testing for Skylab applications.

e. *Applications Guidelines* - The Skylab program uses 27 fans in the cluster ventilation control system and seven fans are kept for spares. No tools are required to remove or replace a defective duct fan. All fans are interchangeable. The fans are operated continuously during the habitation periods and cycled off at the end of the habitation periods.

There are four air compressors for the molecular sieve system on the Skylab program (one spare). Two compressors are used for each molecular sieve; however, only one operates at a time. Although no tools are required for replacement, restraints and tethers are required to allow the use of both hands and to restrain removed items (Ref 8).

The concept used for the Skylab compressors and fans is not to develop long-life components, but to select proven components. Efforts were made to make component replacement an easy task and to have an ample number of these components to last the program duration.

There has not been any inflight maintenance recommended for the pump packages for the Skylab program. The problem was skirted by the use of standby and redundant pump packages.

In view of the previous NASA approaches and the results of the study, it is recommended that the following alternatives be considered)

Mechanical component or module replacements are quite feasible and desirable. System designs, which can accept plug-in or snap-in type items (such as pumps, filters or valves), are recommended.

D. TEST METHODOLOGY AND REQUIREMENTS

The survey of manufacturers and users revealed that accelerated testing techniques generally are not performed on compressors, fans and pumps, however, accelerated testing for these components should be developed. The common qualification tests (applicable to certain units) for these components include: volumetric flow, pressure rise, operating pressure, proof pressure, burst pressure, external leakage, and design life tests. It is recommended that the life tests not be truncated (after an operating time requirement is met), but operated until the unit is out of limits or fails.

A 100-hour run-in prior to flight is recommended by the manufacturers and users contacted for pumps, fans and compressors. It is also recommended that life endurance tests be conducted on those units under operational conditions. These tests would uncover latent manufacturing defects and provide a quantitative judgment of the units life.

One manufacturer has suggested (to extend bearing life) that special bearing inspection be performed to control quantity and contamination of lubricant. It is recommended that the variation of the applied lubricant could be managed by this technique.

In regard to the launch vibration levels expected for the Shuttle Program, the survey of manufacturers and users indicated that these levels should not be a problem to the plain journal bearings used in their pumps. Their experience, has included vibration levels to 10-20 g. The motor bearings would also be vulnerable to these vibration loads, but they can be sized to meet these loadings. The survey reveals that many motors used in engine applications experienced high "g" levels without problems.

Failure Mode Detection - The flight crew may detect a fan failure by audio means or by feeling the fan housing and determining it is not operating. A crewman may then remove power if the fan is inoperative and replacing the defective fan with an onboard spare. The fans are not normally monitored with ground instrumentation.

A coolant pump failure monitor normally activates an associated panel warning light and an audio signal for the flight crew. Telemetered information for both the ground flight and ground crew can include flowmeter readout for the affected loop pump current and start transients, or delta pressure transducer readout which indicates low pump pressure. Since these coolant pumps are normally used redundantly, the appropriate action is to activate the alternate pump.

E. PROCESS CONTROLS

Necessary process controls include the fabrication of bearing material free from inclusions, stringers, or voids. These types of problems cause increased bearing wear and ultimate failure. Manufacturing methods that include a vacuum melt, or even remelt processes, minimize the presence of inclusions and are a recommended practice to improve bearing life.

Housings used in fluid systems must be leak tight. Castings can be impregnated with a sealant to preclude leakage. The housing material (machined) used can include a vacuum melt process to reduce inclusions.

Data from Reference 4 revealed that 22 of 74 bearings in stock for Saturn usage had cracks. It was concluded that the procurement specification pertaining to bearings (high speed, long-life, or heavily loaded) that were used in failure critical applications, should contain a proviso for 100% dye penetrant inspection on balls prior to installation in bearings to detect cracks.

Part Usage Constraints - Compressors, fans and pump life requirements for such programs as Shuttle cannot be met unless maintenance and restoration are permitted. This study indicates that compressors, fans or pumps maximum service life expectancies range up to three years using present state-of-the-art components. Tables 2 and 3 show the advantages and disadvantages of the various components under consideration.

A tradeoff study is needed to evaluate whether a concerted effort need be performed to extend the life of these components, or whether maintenance and restoration can be performed on these components for extended utility.

F. THERMAL CONTROL PUMPS, COMPRESSORS, AND FANS DESCRIPTIVE INFORMATION

1. Thermal Control Pumps

Thermal control pumps, also referred to as recirculatory pumps, are used to circulate a coolant fluid through a heat transport section to control heating in selected electronic packages and the environmental control systems for the Gemini spacecraft, the Apollo capsule, the LM, Skylab equipment and the portable environmental control unit carried in the astronaut backpack.

Thermal control pump packages generally include an electric induction motor, impeller, relief valves, check valves, filters, differential pressure switches, bearings, seals, noise filters, a metal housing (usually machined) and a power source consisting of either a standard ac motor, brushless dc motor, or a solid-state dc/ac inverter motor.

Operational as part of the pump package are such components as accumulators, that provide positive inlet pressure to the pump and provide for fluid thermal expansion.

The coolant fluid media used for aerospace applications include ethylene glycol-water, Freon mix, Coolanol, Freon E or water. Dielectric fluids include Coolanol, FC-75 and Freon, while included in the non-dielectric fluid categories are: water, ethylene glycol, and water methanol.

a. Pumps - The pump can be either centrifugal or a positive displacement rotary (gear or vane).

Centrifugal Pump - These pumps are made up of four major elements; impeller, bearings, seals, and motor/driver. Current centrifugal thermal pumps employ the single-suction impeller type where the coolant liquid enters the suction eye on one side only. These types are being used in the Apollo program.

Gear Pump - They have two or more intermeshing gears or lobed members enclosed in a suitable housing. These are constant delivery, positive-displacement units that are generally used when the selected fluid has high viscosity. The gear pump, because of its compactness, lends itself to tandem or double-pump assemblies. The Gemini and Skylab programs used this type pump.

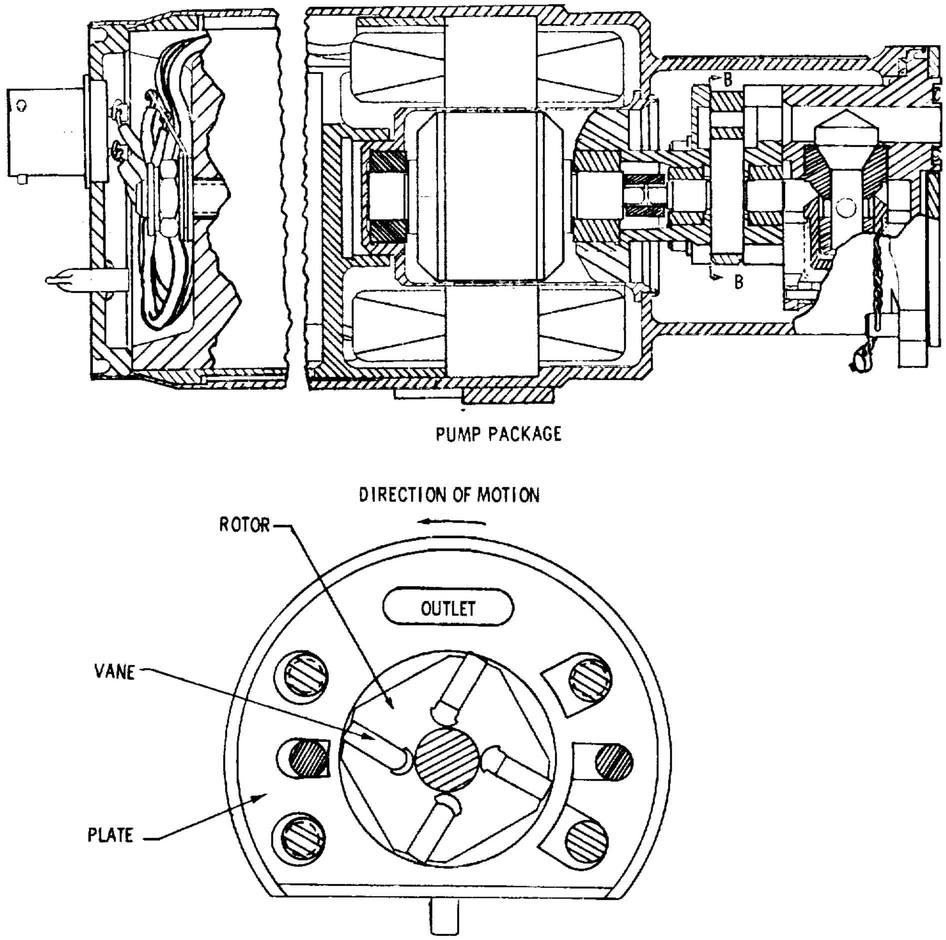
Vane Pump - The type pump may be of either the fixed-delivery or variable delivery type. The former types are power-driven pumps having a constant volume, with multiple vanes within a support rotor encased in a cam ring. The rotor can be stainless steel rotating within a steel liner. To reduce friction, all steel parts; vanes, rotor, lines, and side plates, can be treated with a hard chrome diffusion process. Pumps of this type are presently used in the LM and Skylab coolant packages. Like the gear type, this pump lends itself well to compound element combinations.

The Skylab water pumps (Fig. 4) are positive-displacement, rotary vane, electrically powered pump assembly consisting of five sub-assemblies: (1) pump, (2) relief valve, (3) ac electric motor, (4) dc to ac inverter, and (5) outer housing which encloses the entire assembly. These pumps can continue to operate with the outlet line blocked because the internal relief valve allows flow from the outlet back to the inlet side of the pump when the outlet pressure builds up to the relief valve cracking pressure. Most structural parts are made of corrosion-resisting steel and the bearings are carbon journals. The entire unit is hermetically sealed by welding. The motor stator and inverter are separated from the motor and pump and are sealed in an inert atmosphere.

b. Motor/Drivers - There are three commonly used electric motors in use to drive the pump section: (1) a standard ac induction motor; (2) brushless dc motor; and (3) standard dc motor.

There are numerous motor/driver/coupling combinations. However, for this study only two methods and techniques are described.

Inverter - ac Motor Combination - This driving design has two main components. One includes a solid-state dc/ac inverter that supplies power to the second component; an ac induction motor. The motors are either two- or three-phase ac motors. The special feature of this design was the canned motor stator. The electrical stator was placed in a welded stainless steel "can" to completely isolate the windings from the fluid. An open electrical stator used in a common coolant, such as water/glycol solutions, are destined to fail because of high conductive and corrosive properties of the fluid. The electric inverter employs solid-state electronic circuitry to transform the input dc power to ac power.



SECTION B–B

Figure 4. Rotary Vane Pump

The inverter consists of regenerative external feedback to place the transistors in the saturated mode and also renders the frequency insensitive to voltage and load change. Briefly, the circuit consists of two individual inverters that are cross-coupled with one another in a manner that produces a constant phase shift between them. If reasonably matched, the phase shift will be 90°, as required for the optimum operation of the two-phase induction motor. For the purpose of driving a thermal liquid pump, a nearly constant motor rpm offers the advantage of maintaining a nearly constant flow of fluid, independent of input voltage variations, as well as hydraulic resistance variations. However, when operating at constant frequency and at the lowest input voltage, the motor efficiency may drop significantly and the overload margin may become undesirably small. Therefore, the regulator circuits are adjusted to produce a voltage variation equal to between 1/3 and 1/2 of the input voltage variation. In so doing, the inverter motors are made capable of operating efficiently over a wide range of input voltages and maintain a safe margin of torque capacity at the lowest input voltage. Suitable proportioned controls will maintain a constant starting and pullout torque over an input voltage range of 2 to 1.

Brushless dc Motors - The brushless dc motor design has been proven during testing for the MOL, Apollo, and Skylab programs where Freon-21 coolant was used. The magnet drive coupling principal uses a barium ferrite six-pole permanent magnet mounted on the pump impeller shaft. The magnet rotates in the refrigerant. Inconel covers are placed over the magnet piece and are bolted to the pump housing. The brushless motor that drives the magnet consists of a permanent magnet rotor, ironless stator, a light source and aperature disc (which are used to indicate commutation sequence), and the mechanical parts necessary to enclose the machine, support the bearings and provide the proper mounting pad. Commutation switchings are accomplished in an electric package described in the following paragraph.

There are four primary motor components, and a brief description of each follows:

1) Permanent Magnet Rotor - The rotor magnet use platinum-cobalt. It has a high coercive level and permits operation across large air gaps.

2) Ironless Stator - Because of the high coercive strength of the rotor, no stator teeth are needed to provide a magnetic flux path. This permits use of the full bore area for copper. Stators are made mechanically sound by compacting and epoxy encapsulating the copper, and employing a laminated iron ring around the outside diameter. The ring keeps stray magnetic flux from extending to other regions of the motor. The main stator windings are designed half-wave operations. Secondary windings are placed over the main winding to produce, when rectified, a dc output proportional to speed. Signals are used to monitor the output speed.

Aperature Disc Commutator - A series of matched solid-state light emitters and detectors that are energized when the rotor fields are aligned with a specific stator coil. The light detector signals are used to switch a power amplified that, in turn, excites the specific stator winding with rated direct current voltage.

Mechanical Assembly - The permanent magnet rotors are precision mounted on a shaft. The stators are mounted in housings that permit attachment of end bells. Grease-packed bearings are used. The light emitters and detectors are mounted in the stator housing and aperature discs are mounted on the shafts. This type motor and coupling has proved reliable in many space programs including Apollo and Skylab.

c. Seals - Two types of seals are used in thermal control pumps; static and dynamic.

The dynamic seals are being eliminated by the use of canned motors, dielectric coolant fluids, and the use of magnetic couplings.

Static seals or mechanical seals may differ in various physical respects, but are fundamentally the same in principal. The sealing surfaces on both kinds are located in a plane perpendicular to the shaft. There are two basic seal arrangements; the internal, in which the rotating elements are inside the case and are in contact with the fluids; external, in which the rotating elements are located outside the box.

The pressure of the fluids in the pump tends to force the rotating and stationary faces together in the internal seal, and to force them apart in the external seal.

The main static sealing methods are in use today; flat gaskets, O-rings in grooves, and seals molded in place.

Fans and Compressors - Fans and compressors are similar because they both convert mechanical rotative energy, applied at their shafts, to gas energy.

Most fans and compressors as classified by the geometrical flow path (such as centrifugal or axial). Axial fans are identified by flow within the wheel that is parallel to the shaft. Centrifugal fans have flow within the wheel that is perpendicular to the shaft. Air is drawn into the impeller or fan blades where energy is added, discharged to the case, and forced through the fan or compressors outlet. The energy added appears as an increase in pressure, velocity and temperature.

Figure 5 shows a typical cabin fan used on the Apollo program. The fan is a single-stage, direct motor drive, axial flow fan operated by a 115/120 V, 3-phase, 400 cps power source. The fan was designed to rotate at a nominal speed of 11,000 rpm to function for a 1200-hour minimum operational life.

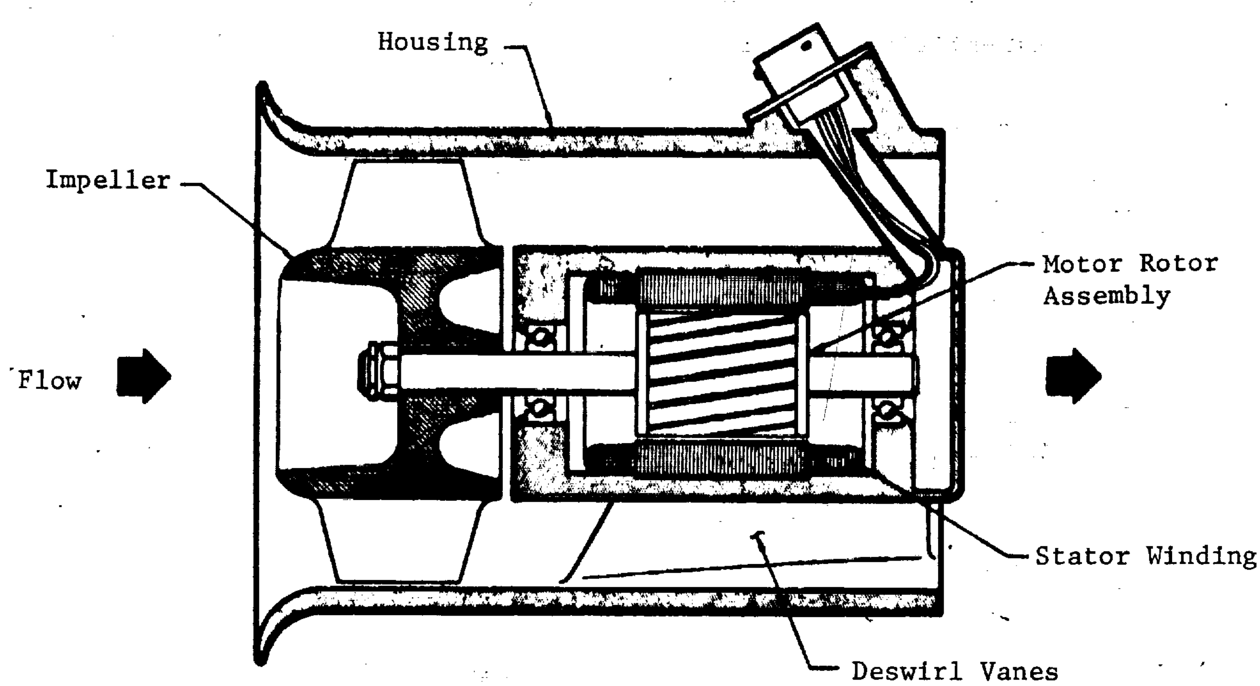


Figure 5. Axial Flow Fan

The molecular sieve compressors (fan) used on the Skylab program are basically the suit compressors that were used in the Apollo CSM, Figure 6. This fan is a single-stage centrifugal design with a single speed and a relatively high flow output with a low delta pressure. Flow through the compressor deflects after passing outward through a diffuser into a conically converging annular passage over the motor housing. The deswirl vanes, which also support the motor, straightens the air flow. An aluminum alloy conduit carries the electrical lead across the flow path into a hermetically sealed receptacle outside the compressor.

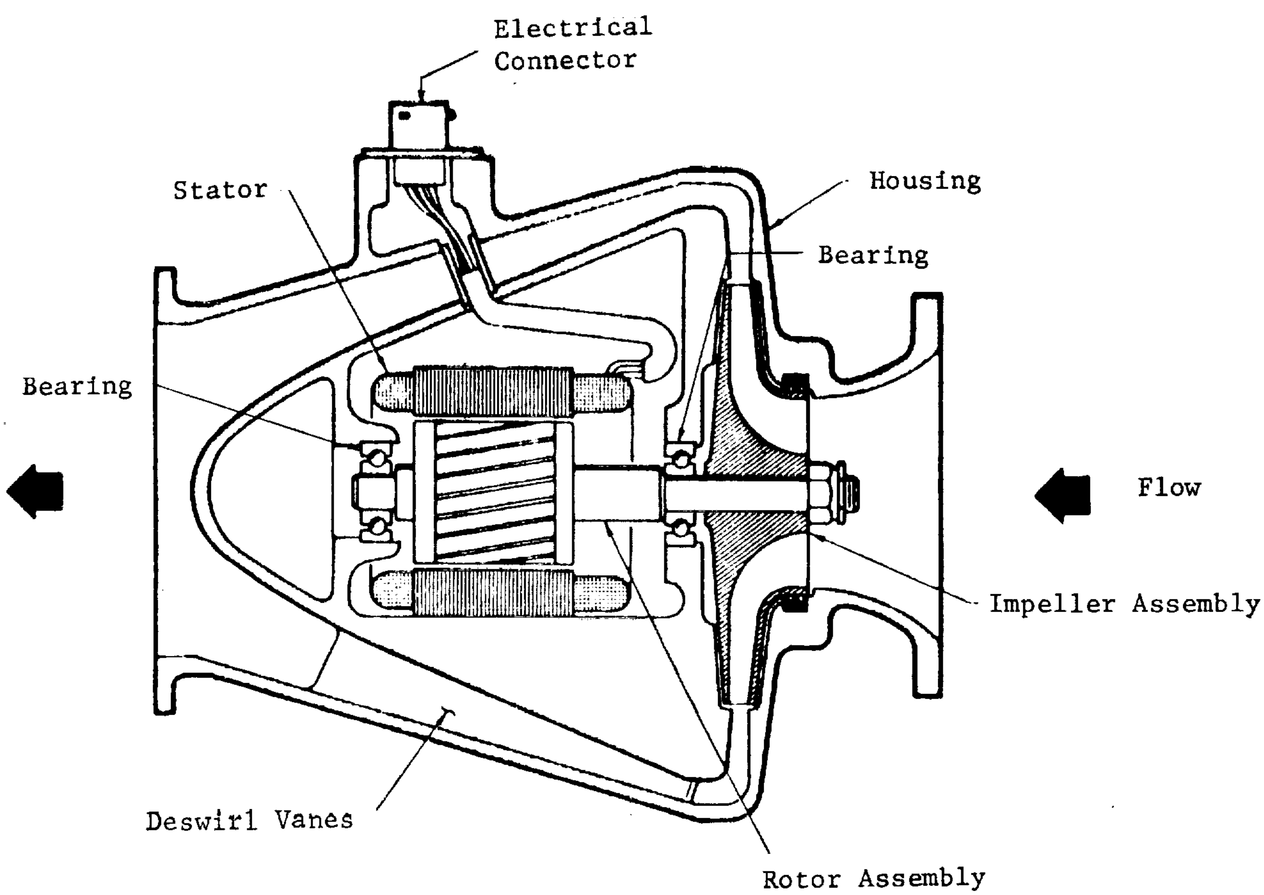


Figure 6. Compressor

G. REFERENCES AND BIBLIOGRAPHY

References

1. Binder, R. C.: *Fluid Mechanics*. Prentice-Hall, 1947

2. DuBuisson, J. C., *et al*: *Handbook of Long-Life Space Vehicle Investigations*. Martin Marietta Corporation, Report No. T-70-48891-012, M-70-16, December 1970.

3. *Design Handbook*. Environmental Engineering, Air Force Systems Command, AFSC DH 1-5, 1968.

4. *Parts, Materials and Processes Experience Summary*. CR114391, Lockheed Missiles and Space Co., Inc., February, 1972.

5. *Aero-Space Applied Thermodynamics Manual*. Published by: Society of Automotive Engineers, 1960.

6. *Reliability Data from In-Flight Spacecraft*. PRC R-1453, Planning Research Corporation, 30 November 1971.

7. Rousseau, J., *et al*: *Environmental Control and Life Support System for Apollo Applications*. Report No. SS-3414, Garrett AiReserach Corporation, 1966.

8. *Skylab Inflight Maintenance (IFM)*. Report No. ED-2002-1237, Martin Marietta Corporation, 1971.

Bibliography

1. Andersen, B. W.: *The Analysis and Design of Pneumatic Systems*. J. Wiley and Sons, 1967.

2. Saltzman, A. R., *et al*: *Design Manual for Methods of Cooling Electronic Equipment*. U.S. Naval Air Development Center, NAVWEPS 16-1-532.

3. Goldstein, A. W., Glaser F. W., and Coats, J. W.: *Acoustic Properties of a Supersonic Fan*. Paper No. 71-182. AIAA 9th Aerospace Sciences Meeting, January 25-27, 1971.

4. Bisson, E. E., and Anderson, W. J.: *Advanced Bearing Technology*. NASA SP-38, 1965.

5. Peterson M. B.: *Literature Search and Critical Review of Mechanical Technology*. Mechanical Technology Inc., Latham, N. Y., July 1970.

6. Butterfield, A. J. and Herz, J. N.: *Experience Applicable to the Viking Lander from a Study of Related Space Flight Projects*. NASA CR-111876, General Electric Company, March 1971.

Persons Contacted

1. Tom Lasker and Russell Perry, McDonnell Douglas Corporation, St. Louis, Missouri (314) 232-7328.

2. Jim Ripple, Bob Morgan and Frank Green, Garrett AiResearch Corporation, Los Angeles, California (213) 323-9500.

3. Dave Rich, Martin Marietta Corporation, Denver, Colorado, (303) 794-5211.

4. D. Bailey, McDonnell Douglas Corporation, Huntington Beach, California (714) 896-2761.

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6. Donald Rethke and Dick Szymanski, Hamilton Standard Corp., East Hartford, Connecticut (203) 623-1621.

VI. MAGNETIC TAPE RECORDERS

by J. C. DuBuisson

VI. MAGNETIC TAPE RECORDERS

A. INTRODUCTION

The history of magnetic recording began with its invention at the turn of the century by Valdemar Poulsen. However, the beginning of large-scale application of magnetic tape recorders did not occur until the early 1950's. By 1966, the April issue of *Instruments and Control Systems Magazine* presented a partial list of aerospace tape recorders and their characteristics consisting of 50 models by 18 companies. Up to 42 channels of data are now being recorded on a single tape. One reason for tape recorder popularity is the high information storage capacity available for the size, weight, power consumption and cost.

The spacecraft tape recorder has three major areas of application. First, they are used for time compression of data; data is read out upon command when the tracking station is available and in position for interrogation. Secondly, they are used for time expansion of data for deep space probes. Data must be recorded at a rapid rate during planet encounter, but must be read back at a much reduced rate because of the bandwidth limitations of the communication link between the spacecraft and earth. Finally, they are used for on-board storage of computer program material for manned spacecraft.

Most spacecraft recorders require multispeed operation because of the time compression or expansion requirements. Multispeed operation can be obtained by numerous methods. Probably the most reliable approach consists of changing the frequency of the power source for the synchronous motor using logic circuitry. The study of electric motors, Chapter II, includes guidelines for electric motors, bearings and lubricant selection for tape recorders.

Very briefly, the magnetic tape recorder functions as follows: Signals from transducers are conditioned and conducted through a coil wound around a magnetic core. A tape with a magnetic oxide coating is passed over a gap in this core; the field induced by the signal creates magnetization patterns on the oxide film. Constraints on information capacity are imposed by noise, linearity or distortion, frequency and mechanical stability. The information is recovered when the tape is passed over a

second head. The tape transport consists of the tape reeling system; tape or capstan drive system; record, reproduce and erase heads; and tape guides. For those desiring more orientation information, suggested reading is "The Elements of the Tape Recorder" chapter of SP 5038.

Magnetic tape recorder signal bandwidth is a function of both mechanical limitations and the relative velocity of the tape with respect to the recording head. Increases in signal bandwidth requirements are met by either increasing the speed of the tape or by increasing relative tape speed by mounting the record/reproduce head in a rotary drum. The record/reproduce head rotates at relatively high rpm's while remaining in contact with the tape, thereby producing relative tape speeds of 1,000 inches per second or more while the tape reeling system handles only up to 120 inches per second of tape.

The life estimates in this chapter are usually expressed in either calendar years or cycles. These estimates assume that the error rate until the end of life is within specifications. Actually three parameters are involved in estimating the life of a magnetic tape recorder, viz: (1) how long (years) the recorder will actually last, (2) the storage rate capability, and (3) the bit error rate at a point in time.

The head may be fixed or rotary. Except for ERTS, only fixed heads are currently employed in satellites because they are less complex, more reliable, weigh less and cost less according to a survey. Rotary head recorders are used primarily for ground television recording with tape speeds up to 1200 inches per second; this type recorder has been used on one earth resources satellite. Because of the lack of spacecraft usage and of timely data of rotary head recorders, only fixed head recorders are studied in this chapter. However, in general, the fixed head recorder discussions apply to rotary head recorders.

The majority of the long-life limiting problems involve the magnetic tape, magnetic head, bearings and drive belts. The electronic long-life problems are not unique to tape recorders and, therefore, not covered in this chapter. Volume II presents some of the long-life problems and solutions for electronics. This chapter places emphasis on the transport elements of the magnetic tape recorder.

B. GUIDELINES FOR LONG-LIFE ASSURANCE

The major life limiting parts of the magnetic tape recorder are the magnetic tape, magnetic head, bearings and drive belts. The solution to the drive belt problem appears to be direct drive design. For manned missions, the magnetic tape wear problem will disappear if the tapes are replaced prior to wearout. A continuously operating tape recorder with a two-year life is within the state of the art. A five year continuously operating recorder can be developed with a low-speed record and medium speed playback. Technology exists to obtain 50,000 tape cycles with optimum tape and transport design.

1. Design Guidelines

1) The materials used for the front face of the head should be harder than Rockwell 100B to reduce headwear, drag and debris generation. Use one of the ferrite alloys such as "Alfesil," "Spinalloy," or "Alfenol."

2) Do not use a section of head cleaning tape on one end of the tape reel to remove debris, glaze and gap smear. Although an apparent short term answer, use of head cleaning tape aggravates wear and subsequent debris formation in long term applications.

3) Employ only instrumentation magnetic tapes and transports with a history of superior performance in spacecraft applications to increase the probability of success.

4) Use NASA/GFSC specification S-715-P-14 which delineates specifications for selecting and using magnetic tape.

5) Limit the maximum stress level in mylar tape to 3000 psi to avoid a region of non-linearity in the region of 8000 to 9000 psi. Keep the tape pack winding tension low to prevent tape deformation.

6) Don't use endless loop transports. They tend to jam or throw loops. Without optimum design a balance between these two failure modes is difficult to maintain.

7) Avoid drive belts which are life limited primarily because of delamination. Use direct drive motors.

8) Minimize the following consistent with constraints to reduce head and tape wear:

a) Tape velocity;

b) Tape wrap angle;

c) Head radius.

9) Consider the merits of the high reliability five-year tape recorder system designed by 11T Research Institute for NASA/ Goddard as a design baseline. They recommended a reel-to-reel coplanar configuration with independently motor driven reels and capstans.

10) Investigate the feasibility of eliminating multispeed requirements by using a semiconductor or magnetic buffer for input/output of data and parallel recording on multiple tracks.

11) Limit the lubricant in the tape binder to the range of 1% to 2% of the total weight of the binder system. Large amounts of lubricant will weaken the binder while small amounts will not reduce the coefficient of friction adequately.

12) Minimize the number and complexity of moving parts to increase reliability.

13) Minimize interacting mechanical function to prevent serially adding functions.

14) Keep the rotational velocity of bearings relatively low to help assure long life. (Too low a rotational velocity will aggravate wear owing to a lack of hydrodynamic lubrication.)

15) Provide bearing lubricant reservoirs to replenish losses.

2. <u>Process Control Guidelines</u>

1) Limit voids or gaps between laminations or other discontinuities on the front face to less than 50 microinches in width. Also, there should be no scratch in the direction of tape motion on the contact surface of the head that is deeper than 12 microinches. There should be no scratches perpendicular to the direction of tape motion. Compliance with the preceeding will reduce the probability of tape damage and decrease the rate of debris accumulation.

2) Cleanliness in assembly is essential to prevent contamination effects. Class 100 clean rooms are suggested.

3) Closely align the gap and apex of the head by observing and maximizing the amplitude of the signal. Misalignment is more critical with the new harder head materials since misalignment cannot be lapped-in from the passage of tape as with soft core heads.

4) Specify and test for the desired tape characteristics rather than attempt to control manufacturing processes since many processes are proprietary.

3. <u>Test Guidelines</u>

1) Thermal cycle head windings and joints six times between -10°C to +66°C to detect open or short failure modes.

2) Subject tape test specimen to the tests outlined in NASA/GSFC specification S-715-P-14 to assure the tape has certain desirable characteristics. These tests include:

a) Thermal stability;

b) Lubricant content;

c) Surface resistivity;

d) Chlorine content;

e) DC noise test (oxide dispersion evaluation);

f) Flexibility test.

4. <u>Application Guidelines</u>

1) Keep the temperature of the tape below 35°C to prevent binder softening and subsequent iron oxide nodule formation.

2) Store tape at 30% RH, at a temperature less than 32.2°C, and in an argon atmosphere to retard aging degradation.

3) Wear-in the head-tape combination to be used in service to reduce tape abrasiveness and eliminate infant mortality. Specification S-715-P-14 recommends 200 passes.

5. Special Considerations

1) Consider continuing development of a fluid filled type of transport that immerses the tape, heads and bearings in lubricant to alleviate many of the wear and lubricant depletion problems. Emphasis should be placed on increasing the recording density (bits/inch) to that of other recorders.

2) An additional study of alternatives to magnetic tape recorders should be conducted to ascertain optimum data storage/retrieval systems for specific missions. Alternatives should include optical/thermal, magnetic disc, silicon drum, dynamic MOS RAM's and magnetic bubble techniques.

3) Consider, for manned missions, the Utility Modular Maintenance concept as a limited-life-component-replacement-methodology to reduce replacement time and installation error probability.

4) Use digital recording if constraints permit. Digital recording permits saturation recording and re-clocking to remove jitter.

5) Investigate the feasibility of air bearings for continuously operating recorders. Because of start-stop wear, air bearings are not suitable for intermittent operation.

6) Determine the areas of airborne aerospace applications of rotary head recorders. The data from the ERTS program should be useful.

C. LIFE LIMITING PROBLEMS AND SOLUTIONS

All agencies and manufacturers interviewed agreed that the major life-limiting parts of tape recorders were magnetic tape, magnetic head, bearings, and drive belts. Some rate these problem areas differently with respect to magnitude. However, they agree that all were important from a research and development standpoint if tape recorders were going to maintain pace with long-life mission requirements. The solution to the drive belt problem appears to be direct drive design. For manned missions, the magnetic tape wear problem will disappear if the tapes are replaced prior to wearout.

As indicated in the introduction, the electrical/electronic long-life problems are not unique to tape recorders and are not covered in this chapter. Volume II covers various aspects of electrical/electronic problem areas and solutions. The electronics of a magnetic tape recorder must conform only to the electronic requirements of other typical electronic equipment for similar environments. Studies of tape recorder failure modes reveal that the number of failure modes for the mechanical tape transport section far outnumbers those for the electronic system (Reference JPL).

Multi-track recorders complicate the electronics, but does not change the basic long-life problems and solutions. However, the greater the number of tracks, the greater the effects of the anomalies. Multi-track recorders (to 42 tracks) increase electronic design problems such as cross-talk, but life problems remain essentially the same. Crosstalk requirements are more severe for an analog than digital recording. Investigation and discussion of basic design problems, not related to life, is not within the scope of this study.

1. Failure Mechanism Analysis (FMA)

Table 1 presents a summary of the failure mechanism analyses. It delineates the failure modes of mechanical components, magnetic tape and the case; their effects upon the recorder; the probable failure mechanisms; detection methods; and recommendations on how to eliminate/minimize the failure modes. The following paragraphs discuss and elaborate upon Table 1. In practice, the failure modes involving the head/tape interface must be considered as an entity; however, for convenience, the head and tape failure modes are discussed separately.

a. Magnetic Heads - Two main types of reproduce heads and one type of record head are in current use. Most heads use the fringing effect that occurs at a well defined gap in a soft structure as the primary mechanism of either inducing magneto-motive force into the tape or extracting flux from the tape. The "d ϕ/dt" and flux sensitive heads are the two kinds of reproduce heads. The d ϕ/dt head is almost identical in construction to the record head. When the rate of movement of the tape is extremely slow, the d ϕ/dt output is low and the flux sensing heads are usually employed. The reproduce heads usually have much smaller gaps than record heads since the width of gap (within limits) does not effect the overall resolution in the recording heads. Recording head gaps are usually in the range of 0.5 to 2 mils. Reproduce head gaps are sometimes as small as 20 microinches. In addition to record/reproduce, it may be necessary to perform the function of erasure of the tape. In pulse recording, erasure is not usually required since the recording current usually saturates the tape and previous tape history has only a minor effect on the record left on the tape by a saturating signal. Direct analog or FM requires an erasure function. The best methods is to erase tape in bulk by passing the tape through a saturating ac magnetic field which is slowly reduced to zero, leaving the tape in an essentially unmagnetized condition. However, the airborne erase function is normally performed by applying a strong high frequency current to a head; this requires large amounts of power. The erase function can be accomplished without the use of power by using a three-stage permanent-magnet erase head. OGO and Nimbus satellites used this approach.

Table 1 Failure Mechanism Analysis-Magnetic Tape Recorders

Part & Function	Failure Mode	Rel. Rank[1]	Failure Mechanism	Detection Method	How to Eliminate/ Minimize Failure Mode
a) Heads (Conduct Signal)	Wear	2	High Tape Speeds & High Footage Passed	Not applicable - a design decision	Use lowest tape footage and speed consistent with design constraints. Select a tape with high resolution, allowing lower tape speed. Employ head material at least as hard as Rockwell 100 B.
			High Tape Contact Pressure (tension)	Not applicable - a design decision	Use lowest tape tension, wrap angle and head radius consistent with assuring tape-head contact, adequate capstan drive function, etc.
			Abrasive Tape	Pre-service tests for abrasion characteristics	Wear-in tape for 50-200 cycles. Procure less abrasive tape, other parameters being satisfactory.
			Corrosion	Reduced signal output/input	Purge transport with dry inert gas. Use compatible lubes.
	Contamination		Tape Wear Products	Intermittent or loss of recording/replay signals.	Use a harder finished tape such as Pyrotrack.
			Lubrication Migration	Intermittent or loss of recording/replay signals.	Proper seals. Minimize amount of lubricant on bearings. Use a lubricant compatible with head material.
			Epoxy Outgassing	Intermittent or loss of recording/replay signals.	Keep temperature below 50°C. Separate electrical system from mechanical system.

[1] 1 = most probable failure mode.
2 = next most probable, etc.

Table 1 (cont)

Part & Function	Failure Mode	Rel. Rank	Failure Mechanism	Detection Method	How to Eliminate/ Minimize Failure Mode
Heads	Magnetic Open		Gap Expansion	Intermittent or loss of recording/replay signal	Due to temperatures above about 80°C. Maintain temperature below 50°C, to prevent open and deterioration of electronics.
	Magnetic Short		Gap Smear	Signal distortion	Caused by contact stress fatigue. Plate pole tips with hard material.
					Pass abrasive tape across to remove or resurface head.
			Winding and Joint Defects	Temperature cycling acceptance test	Tighter manufacturing control.
	Winding Short or Open		Manufacturing	Continuity	Thermal cycling (-25°C to 65.6°C) prior to continuity checks.
b) Tape (Record Signal)	Irregular Surface (nodules)	1	Iron Oxide Wear-Off and Accumulation	Visual examination of tape. Intermittent or loss of signal. Bit dropping.	Use harder surfaced tape such as Pyrotrack. Heads should be smooth and gap aligned to prevent tape damage & debris buildup.
			Binder Softening, oxide builds up	Intermittent or loss of signal	Keep tape below 50°C (35°C preferred) for mylar based tapes. Test tape for adhesion or oxide crumbling characteristics at 175°C.

Table 1 (cont)

Parts & Function	Failure Mode	Rel. Rank	Failure Mechanism	Detection Method	How to Eliminate/ Minimize Failure Mode
Tape (cont.)	Break or Deformation		Start-stop loads excessive	Loss of signal, flutter	Proper design.
			Defective tape or insufficient tensile strength		Derate tensile strength of tape (Mylar 3000 psi max.) H-film is stronger at elevated temperatures. Keep temperature below 50°C.
			High winding stresses on reel.		Proper design. Packing tension must be low.
	Wearout		High speeds, footage and tensions	Intermittent or loss of signal	Use reel-to-reel configuration. Reduce number of tape guides, guide on back side.
	Jamming		G-loads when recorder non-operative	No signal - no rpm	Avoid endless loop transports and clutchless design.
c) Bearings (Support rotating load)	High Starting and running torque	3	Lubricant depletion and deterioration	Slow starts and high motor currents	Select stable lubricant (see Chapter II). Keep rpm down. Employ lubricant reservoirs or equivalent. Consider fluid filled transports.
			Contamination	Visual examination prior to sealing case	Employ usual clean assembly techniques. Assemble on laminar flow clean work bench. Case pressurization gas should be filtered and inert.

Table 1 (concl)

Part & Function	Failure Mode	Rel. Rank	Failure Mechanism	Detection Method	How to Eliminate/ Minimize Failure Mode
Bearings (cont)	Brinelling Pitting		Vibration	Erratic Torque requirements	Operate recorder during periods of high vibration.
d) Drive Belts	Break	4	Delamination	No tape movement	Use direct drive electric motors. Correct design-large radii pulleys, low ft/sec. Use H-film material.
	Slip		Delamination, creep, low belt tension	Erratic tape movement	Use direct drive. Correct design.
e) Case	Leakage	5	Poor workmanship, nicks, scratches, improper o-ring installation	Loss of case pressure	Improved Q.C.

As indicated in Table 1, wear is one of the major failure modes of the head. The wearout characteristics of the magnetic head are determined primarily by the feet of the magnetic tape passing over the head, tape tension, tape contact angle, abrasiveness of tape, and head materials. Head wear is acute in applications where the head dimensions are small and the head-to-tape pressures and velocities (total footage) are comparatively large. The end of the useful life of a head occurs when the tape wears through the depth of the gap and the gap opens up.

Head wear can be reduced by passing the lowest footage of tape at the lowest tension consistent with design constraints such as packing density, resolution, frequency response and percent of flutter. It is not possible to place absolute limitations on tape speed and tension since trade-offs for particular applications are required. A typical tape tension is one pound for a one inch wide tape. Common tape speeds vary from less than one inch per second (ips) to well over 100 ips. Video recorder velocities run as high as 1500 ips.

Improved tapes, such as 3M's 971 high energy series, show promise of high resolution and, hence, lower tape speeds. "Spinalloy" has created some interest in that it is claimed that heads made with ferrite cores, tipped with laminated Spinalloy, can operate at one-half to one-quarter the tape speed normal with other present heads. This alloy developed by Spin Physics of San Diego, California, is a four-element magnetic alloy with high permeability and a Vickers hardness greater than 600. Resolutions of 20,000 to 30,000 sine waves per inch are reported (Reference 1). Spin Physics Inc. claims a head life in excess of 1000 hours at 120 ips. The higher the data packing density, the fewer the tape passes required for a specific mission.

The need for an improved head material is indicated by relatively low wear lives. For example, Mr. J. Baird, White Sands Missile Range, stated they were getting about 500 hours of head life when recording at 120 ips. Mr. J. Allison, VAFB, stated their recorder heads only last 250 to 400 hours (Reference 2). In addition to degraded recording characteristics, a worn head can shred or damage the tape.

Some manufacturers are using ferrite to make magnetic heads. Ferrite heads are becoming the state of the art according to Mr. D. Bixler of 3M Company. In 1960, RCA developed an aluminum-iron-silicon alloy metal pole tip known as "Alfesil," which is

used for commercial video head wheel panels for operations up to 1560 ips. The present state of the art of aluminum heads without Alfesil tips is 0.002 inch of wear in 5000 hours of operation at 15 ips (verified by test data). Extrapolation of wear test data using Alfesil tips, indicated that head life could be increased by a factor of about three if Alfesil tips were employed. Alfesil heads used for earth resources spacecraft program lasted ten times longer than conventional materials according to Reference 3.

Another problem is contamination (debris) buildup on the heads. Also, a glaze can build up on the head. These problems result in intermittent or loss of recording signals. Most contamination is from the iron oxide coating on the Mylar tape. Abrasive resistant additives and lubrication of the tape can considerably reduce this problem. However, the lubricant may cause its own problems, as explained in the next section on tapes. The use of a section of head-cleaning tape on one end of the tape reel can be used to remove some of the head contamination and glaze for short term applications. Recent re-examination of the use of head-cleaning tape indicates that its use aggravates wear and subsequent debris formation. Pyrotrack magnetic tape, due to its extreme hardness and smoothness, shows promise of reducing this problem.

Testing by IITRI (Reference 4) indicated head material was a factor in establishing the functional drag characteristics of a head/tape combination. It also indicated that the condition of the front surface of the heads was an important factor in the generation of debris regardless of the tape used. Since many adhesion failures are preceded by excessive buildup of debris on the heads, some interrelationship exists between the two areas. To reduce drag, debris generation and head wear, the materials used for the front face of the head should be hard--greater than 100 Rockwell B. Brass should be avoided. Alfenol and Alfesil are among the preferable magnetic materials. Chrome, stainless steel, and Havar are among the preferable non-magnetic materials.

Discontinuity or voids in the head can become a trap for a particle. This, in turn, can cause tape damage and an accelerated rate of debris accumulation. It is recommended that voids or gaps between laminations or other discontinuities on the front face should be less than 50 microinches in width as suggested in Reference 4. Also, there should be no scratch in the direction of tape motion on the contact surface of the head that is deeper than 12 microinches. There should be no scratches perpendicular to the direction of tape motion.

Another type of head contamination failure was detected during temperature qualification testing (above 50°C) of transports at NASA/Goddard. These failures were caused by outgassing of epoxies. This problem was solved by separating the electrical system from the mechanical system in the recorder.

Head problems have also occurred in temperatures above 80°C with expansion of the gap that results in an open failure. One corporation has developed a proprietary fix. This fix has been tested successfully on a single-channel head in a thermal environment of 175°C.

Gap smear occurs when the core material flows across the gap causing a magnetic short. The flow is believed to be caused by contact stress fatigue. The remedial action is to resurface the head (if possible). Smear effects can also be reduced by using harder materials such as Spinalloy and using widest gap possible.

Open and short failure modes results from manufacturing defects can also occur in the windings and joints of the recorder head. Tight manufacturing specifications and temperature cycling acceptance testing (-10°C to +66°C) have virtually eliminated this problem. See Chapter II of Volume IV for a thermal cycling analysis.

Carroll and Gotham (Reference 5) found that the presence of humidity at the head/tape area causes a large increase in wear; almost proportional to the relative humidity. The humidity condition of the tape, or tape surface, was found to have very little effect on wear. The wear rate was not the same for all head materials, indicating the presence of moisture caused corrosive wear. It is suggested that the transport section be charged with dry nitrogen or equivalent if the head material responds to humidity.

b. Magnetic Tapes - The consensus of the general industry and users survey indicated the magnetic tape to be the major life-limiting factor of spacecraft magnetic tape recorders. A tape failure is defined as any breakage, wearout, or deformation of the tape that will cause inability to retrieve data in a form which will allow meaningful interpretation. Useful tape life varies considerably because of tape quality, transport design, environments, tape speed, and operating time requirements.

Magnetic tape life is limited by two basic failure mechanisms. First are the *fatigue type failures* resulting from passage of the tape at high speed over small radius capstans, rollers, or record/

reproduce heads and the presence of excessive tension in the tape base material. Secnd are the magnetic recording medium failures resulting from *chemical or thermal reaction* and abrasion produced by particular contamination.

Three magnetic tape manufacturers were interviewed regarding tape life. Table 2 identifies the tape manufacturers interviewed, the type of tape manufactured by each, and the advantages and disadvantages of each type of tape.

Tape manufacturers are frequently unable to explain why the quality of tape varies. Two recorder manufacturers have developed rigid specifications for selecting quality tape. Using these specifications, only 30% of the tape is acceptable, increasing tape costs. It is suggested that only instrumentation tapes with a history of superior performance in spacecraft applications be employed. NASA/GFSC S-715-P-14 (Reference 6) delineates specifictions for selecting and using magnetic tapes.

There is no generally accepted formula to compute either the life or the reliability of tape. Testing has revealed a wide variance in tape life between transport designs. Some worse-case transports with high density, high tension, and low dropped bit rate requirements are limited to 500 tape passes. Other machines with low tension transports, designed specifically for long-life have operated for more than four years, logged approximately 40,000 cycles on its tape. Raymond Engineering logged more than 25,000 tape cycles on a spacecraft recorder during laboratory long-life testing.

Both the Tiros 8 and the Raymond test recorder are classified as low-speed record/medium-speed playback recorders. The industry generally believes that the state-of-the-art exists in transport design capability and tape development to achieve 50,000 tape cycles.

The solution to the problem of achieving long-life magnetic tape is lagging behind the development of other recorder problem components.

IIT Research Institute is currently investigating the problems of long life tape under NASA Contract NAS5-21623; a five-year magnetic tape is being formulated specifically for unattended satellite tape recorders.

Table 2 Status of Long-Life Magnetic Tape

Manufacturer (Type Tape)	Advantages	Disadvantages
3M Company (Iron oxide coating bound on Mylar)	Proven tape life of 50,000 tape cycles with short length of of tape on low tension transport.	Quality of tape not consistent; Binder and Mylar softens when subjected to temperatures over 50°C, consequently, iron oxide separates from Mylar; Not sterilizable.
U.S. Magnetic Tape Co. (Iron oxide coating bound on (Kapton) H-film)	Has an operating capability in a temperature range from -55 to +100°C; Tensile strength of H-film predicted to be 10 times greater than Mylar.	Reliability and life unknown; Estimated to be not sterilizable. 4 to 5 times better than Mylar.
Lash Laboratories (Stainless steel)	Sterilizable; Long tape life.	Requires higher tape tension and precise head alignment to maintain proper head to tape contact. Consequently, head wear is greatly accelerated.
Lash Laboratories (Copper film sandwiched between cobalt-nickle plating and (Kapton)H-film; name of tape is Pyrotrack)	Has operating capability in a temperature range from -55 to 250°C (sterilizable); Life predicted to be longer than that of transport due to hardness, and playability of tape; Low head wear due to low tension operating capability and use of solid film polymer or silicon lubrication.	Life and reliability not substantiated by testing; Wrinkling on edges occurs if tape tension is high; Average bit drop of approximately 6 bits/600 ft of tape; Not recommended for use in a center pullout endless loop recorder.
Lash Laboratories (Silver lubricated)	Proven at 8,000 passes. Low tape/head wear due to smoothness.	Speciality item. Little actual use history. Lubricant migration during long storage periods.

The major failure mode of magnetic tapes is the formation of nodules on the tape. These nodules are usually from accumulations of iron oxide. These irregularities can cause intermittent loss of signal or bit dropping. A harder surface tape and a smoother magnetic head can alleviate the iron oxide formations. Also, keep the temperature of the tape below about 50°C. With most Mylar base tapes, temperatures have to be maintained below 50°C to prevent softening of the blinder, causing the iron oxide to separate from the Mylar. A maximum temperature of 35°C is suggested to provide a safety factor. High temperatures also cause a softening and loss of tensile strength of the Mylar base material. This has not proven to be a difficult constraint on past spacecraft programs because the thermal control systems were designed to maintain the temperatures below 50°C. However, the Viking sterilization qualification requirement is a much more stringent constraint than was previously required. Table 2 identifies the magnetic tape that can withstand this environment.

The stress levels to which the tape is subjected will ultimately affect the debris and dropout performance of the overall system. Measurements of the stress-strain characteristics of the tape indicates a region of nonlinearity exists in the region of 8550 psi. The tape should not be stressed to the extent that operation in this region is encountered. The maximum stress level in the tape should not exceed about 3000 psi at any point in the tape path.

Tight winding of the tape pack can produce severe deformation even at room temperature. Therefore, packing tension must be low.

The abrasiveness of the tape affects head wear. It is evident that a tape brand with low abrasive qualities should be chosen, other parameters being equal. In addition to tape brand selection, "wearing-in" the tape will reduce head wear. In 1966, Carroll and Gotham (Reference 5) performed abrasion tests utilizing metal rods, 1/2 inch in diameter of various materials such as Hy-Mu 80. After passing several thousand feet, the test rod wear was determined. The results indicated two things: the first pass is the most abrasive and the abrasivity decreases with the number of passes--just as sandpaper wears out with usage. Data indicates that over fifty passes of the tape over a simulated head will reduce the abrasivity to a satisfactory level. Therefore, as recommended in Section D of this chapter, tapes intended for long-life missions should be worn-in for 200 cycles. After the wear-in clean the tape with a commercial cleaner.

Spacecraft recorders for the Orbiting Solar Observatory (OSO) experienced tape problems during qualification testing on two types of transport designs. The problems were caused by g-loads during non-operational shock and vibration acceleration tests; viz:

1) Clutchless transports experienced problems with the tape reeling off the reel.

2) Endless loop transports (not recommended for long-life application) experienced problems with the tape jamming.

These problems were solved by operating the recorders during periods of high g-loading.

c. Bearings - The long-life problems of bearings in a space environment are not particularly unique to magnetic tape recorders. The problems and solutions are more fully defined and discussed in Chapters II and IV of Volume III, concerning gyroscopes, electric motors, and bearings. This chapter shall only briefly discuss bearings and their failure mechanisms. The factors that limit the life of bearings are:

1) Lubrication depletion and deterioration.

2) Excessive contamination.

3) Inadequate quality control selection techniques.

4) Improper transport design.

Lubrication depletion and deterioration is by far the most prevalent cause of bearing failure. This is because only very small amounts of lubricant can be used on recorder bearings; lubricant is limited to prevent contamination of other parts of the recorder via lubricant migration. When bearings are analyzed after several months of operation, breakdown products initially appear as wet grease-life particles, later as crunchy brown particles, and finally as hard cinder-like particles. This polymerization failure mode can be controlled by selecting a stable lubricant. However, selection of the best lubricant for a long-life is difficult because there are no hard facts regarding life test data. (Andoc C is gaining general acceptance.)

Second to lubricant deterioration and depletion, contamination causes the most bearing failures. Most failures occur in the early stages of spacecraft recorder research and development. Contamination failures have been considerably reduced since most spacecraft recorder manufacturers modified their cleaning facilities, assembly facilities, and test inspection techniques to be compatible with spacecraft contamination environment requirements. Most recorder manufacturers maintain their hardware to a Level C cleanliness requirement and assembly facilities are maintained at a Class 100 clean room requirement. Once the units are assembled, hermetically sealed, and pressurized with a dry gas, contamination ceases to be a problem as long as the seal is maintained and the thermal environments are controlled.

d. Drive Belts - All recorder manufacturers surveyed who employ drive belts, use endless loop polyester (Mylar) belts in the speed variation and power transmission of production units. Mylar belts were originally developed by Light and White (NASA TN D-663, May, 1961). The Mylar belts are made by cutting a "donut" from 0.001-inch thick bar stock, stretching it on a mandrel, and heat-treating for stress relief. Leach testing of belts at 24,000 rpm emphasized that material selection, forming, and heat-treating processes play a major part in increasing belt life. Delamination of the Mylar is the most prevalent cause of belt failure in long-life applications. Delamination is the result of greater tension being placed on the outer layers of Mylar than on the inner layers. Consequently, unilateral belt fatigue starts on the outer Mylar layers and migrates to the inner layers. Use large diameter pulleys to increase life. W. Van Keueren and E. Cuddihy of JPL have recently carried out extensive investigations concerning belt drives.

The best way to solve the drive belt long-life problem is to circumvent the problem by not using belt drive. Use direct drive motors. Utilizing direct drive servo-controlled motors, transports with speed ranges of greater than 200:1 are now possible without "cogging" problems at low speed (Reference 1). Hysteresis synchronous motors are used in most present tape recorders. Some use a brush dc type. The latest trend is to eliminate belts by employing dc brushless motors with a servo speed control loop; despite the additional control complexity,this approach is favored in many quarters. Some of those interviewed preferred the less complex hysteresis synchronous motorpole (polyphase) even though drive belts are required; this configuration is acceptable where there is little speed variation. We suggest the brushless dc motor where a wide speed range is required. See Chapter II for a discussion on electric motors, including motors for tape recorders.

e. Recorder Case - Recorder enclosure seals caused the highest percentage of problems during testing of Tiros, Nimbus, Mariner, and OGO satellite programs. Most of these failures were the result of quality control problems experienced by one recorder manufacturer. These failures were due to damages or improperly installed O-rings, nicks, scratches, or burrs in the casting sealing surfaces, defective fill valves, and damaged electrical pins that broke sealing surfaces. This is not considered a recorder long-life problem that is beyond the state-of-the-art, but is a quality control problem that can be corrected.

The state-of-the-art for sealing tape recorders has been proven on the Titan program where pressure transducers were installed on 13 satellite tape recorders. On these 13 satellites, more than 140,000 hours have been logged with only two sealed units dropping below the required pressure. The two dropped below the pressure requirements after 10,000 and 13,000 hours of flight operation.

No environmental problems are anticipated with sealing if the case is designed so softwear sealant (O-rings, epoxies, sealing compounds) exposure to the hard vacuum atmosphere is minimized.

2. Design

a. Selection Criteria - System design is of paramount importance if recorder long-life is to be achieved. Table 3 presents the important design considerations in selecting a recorder. Some aspects of design have been discussed previously. Further elaboration is contained in this subsection to further clarify some of the design factors of Table 3.

Tapes - Use the lowest permissible tape velocity consistent with performance requirements. Actually, the goal is to pass as few feet of tape across the head for a given mission as possible to reduce tape and head wear. The general consensus was that 1 to 30 ips is normally optimum. Above 30 ips the tape starts to lift off the head and more tension is required to assure intimate tape-to-head contact. Although some recorders operate satisfactorily below 1 ips, the following undesirable characteristics occur: obvious wear increase, reduced S/N ratio and the flutter percent is higher.

The head-to-tape pressure (Design Factor 2) should be the minimum consistent with intimate tape-to-head contact. The required head pressure for a given tape is a function of tape tension on the low tension side of the head, wrap angle and head radius. Equations governing the interaction of these factors indicate the following:

Table 3 Design Factors for the Long-Life Assurance of Magnetic Tape Recorders

Design Factors	Remarks
1. Tape Speeds	Use the lowest permissible tape speeds consistent with performance characteristics such as resolution. Determine which vendor can provide the necessary recording characteristics at the lowest tape speed. Select a magnetic tape with the best resolution characteristic, other parameters being satisfactory; a high energy tape such as 3M's 971 tape may allow slower tape speeds.
2. Head-to-Tape Pressure	The design should employ the lowest uniform head-to-tape pressure consistent with intimate tape-to-head contact. The required head pressure will increase with tape speed because the tape will "fly" away from head.
3. Magnetic Head Materials	The head material should be one of the newer harder materials with good resolution and bandwidth. Ferrite heads are preferred. Consider "alfesil", "alfenol", or "spinalloy".
4. Tape Transfer	Use reel-to-reel rather than continuous loop which tends to jam. Avoid tape guides to reduce wear. Guides should rub on off oxide side. The only moving contact (slippage) should be with the tape recording head. Keep maximum stress level in the tape below 3000 psi at any point in the tape path. Crown rollers are superior to edge guided rollers.
5. Tape Material	The tape selected should be relatively non abrasive to reduce head wear. See text for selection criteria.
6. Transport Complexity	The design should use the minimum number of moving parts. An isolastic drive system, for example, eliminates about one-half the working parts. A Newell reel drive is relatively simple.
7. Belts	Designs should avoid their use. Employ direct motor drive with servo-speed control.
8. Bearings	Employ the minimum number possible. The required rpm's should be low. A lubricant reservoir, or equivalent, should be incorporated into the design. See Chapter II for lubrication and bearing details.
9. Transport Alternatives	For unmanned missions of greater than five years duration, the fluid filled transport is recommended for consideration. See reference 4 for a design study of a five year spacecraft tape transport.
10. Input Signal	A digital input is preferred over an analog input because a saturation signal can be applied to the tape with a digital input.

1) Tape wrap angle should be minimized;

2) The minimum head radius consistent with the packing density of the information being stored should be utilized.

The reel-to-reel recorder is considered a better design to accomplish long-life than the endless loop recorder, even though the reel-to-reel recorder requires more electric control circuitry. This opinion is based on the fact that the tape on endless loop recorders requires more lubrication to achieve long-life. It is also susceptible to jamming or throwing loops. The real-to-reel recorder tape can achieve a relatively long-life without lubrication.

As indicated previously, the quality of magnetic tapes can vary considerably. Hence, tape selection criteria was developed by IIT Research Institute for NASA/GSFC. The criteria includes thermal stability, lubricant content, resistively, chlorine content, oxide dispersion and flexibility. The criteria (Reference 6) is presented below:

1) There should be no evidence of adhesion or oxide crumbling after a prescribed thermal stability test of tape test specimen at 175°C for five seconds. See the test section (D) for a description of the test. Lack of thermal stability can allow softening and deposition of melted or plasticized debris on the head;

2) The lubricant contained in the binder should be between 1% and 2% of the total weight of the binder system. The lower limit of percent lubricant content is necessary to obtain the desired reduction in coefficient of friction, while excessive quantities are believed to be a factor in weakening the integrity of the binder polymer system and causing transport slippage. The most prevalent lubricant is organo-polysiloxanes;

3) The surface resistivity of the tape should be greater than 0.5 X 10^7 ohms and less than 50 X 10^7 ohms when measured at 30% RH and 22°C. The surface resistance on the oxide side is related to the amount of carbon or graphite added to the binder. Excessive carbon can weaken the binder;

4) Tapes containing chlorine should not be used in applications where the tape is expected to remain stationary against the head under tension for long periods of time. Tapes containing chlorine exhibit increased adhesive interaction with various head materials;

5) The tape noise reproduced at 30 ips from a tape magnetized to saturation in one direction should be at least 63 db below the saturated signal at 15 kHz over a 10 kHz total bandwidth. This dc noise test indicates oxide dispersion, although sensitive to oxide surface finish and mylar thickness variations. Oxide dispersion can also be evaluated by examining cross-sections of tape using a transmission electron microscope. Tapes having a uniform dispersion of the magnetic oxide, i.e., without agglomeration, tend to produce less debris with usage;

6) The tape should have a minimum flexibility. The deflection of three inches of tape from the horizontal should not be less than either 30 degrees for a one mil tape or 35 degrees for a 1.5 mil tape ½ inch wide. The more flexible tapes are less likely to generate debris.

Transports - As indicated in Item 6 of Table 3, design for the minimum number of moving parts. For example, the Kinelogic Company designed a transport with an isolastic drive system. The advantages of the system are:

1) Approximately half of the working parts are eliminated.

2) No slippage.

3) No energy dissipation. Little energy required by the take-up reel. Energy is furnished by the supplied reel.

The disadvantage is that all of the stresses of the drive system are exerted on one drive belt.

Proper transport design is necessary to achieve low wear/long-life of heads. This was illustrated by testing of heads of two transports. One transport design was operated for 5000 hours at 15 ips with only 0.002-inch gap wear, which is considered near optimum performance for aluminum heads. The second transport design was operated for 200 hours at 15 ips and showed a 0.006-inch head wear. The extreme difference in head wear between these two transports was caused by the higher tape tension and the more severe abrasiveness of the tape on the latter transport. The head materials were the same. These tests illustrated the part that proper transport design plays toward achieving long life of magnetic heads.

The IIT Research Institute has conducted a number of long life tape recorder studies for both NASA Goddard and industry. Reference 4, "Design Study of a High Reliability Five Year Spacecraft Tape Transport" is suggested reading.

In this study, a trade-off study was performed between five reel-to-reel transport configurations. The study indicated the reel-to-reel coplanar configuration with independently motor driven reels and capstans as the most desirable candidate for a high reliability five-year tape transport. A design study and layout of the winning configuration was then performed by IITRI. The design study included bearings and lubrication, tape management system, testing techniques, etc. It should be noted that a parameter "weighting" approach was used in selecting the winning configuration; not all those interviewed agreed with the weighting factors.

One of the losing candidates in the IITRI study was a coplanar Newell reel drive. A Newell transport is being tested by NASA/MSFC. The Newell principle is a mechanical technique for rolling tape from the tape roll rather than pulling the tape as conventional rollers do. This is a relatively simple drive with low power consumption. It was rated low primarily because of lack of proven applications.

Drive Belts - Avoid the use of belts because of their relatively limited lives. Most of those interviewed recommended direct-drive servo-controlled dc motors, either brush or brushless. If belts cannot be avoided, most tape recorder manufacturers believe that optimization of belt drive design can extend the life of belts beyond the current life of 10^9 cycles. Some of the design aspects that should be considered are the use of larger pulleys, longer belts, lower speed drive motors, and other speed reduction devices. However, most of these changes will result in a weight and volume increase.

The most optimistic development in extending belt life is the development of polyimide (Kapton, H-film). Two recorder manufacturers stated that testing on polyimide belts has indicated an order of magnitude increase in the life over polyester belts. Some tape recorder manufacturers feel that with an improved transport design and use of polyimide drive belts, the life of belts can be extended from 10^9 to between 10^{10} and 10^{11} belt cycles.

Bearings and Lubrication - The long-life problems of bearings in a space environment are not particularly unique to magnetic tape recorders. Because of spacecraft power limitations, the power available to the electric drive motors is usually limited, limiting torque margins of safety. Any variations or increase in starting or running torque requirements due to bearing or other problems could create problems associated with varying tape speeds and tensions such as flutter.

Keep the rotational velocity of the bearing relatively low. High-speed operation of bearings will accelerate depletion of lubrication. This is because the heat generated accelerates the evaporation process of the lubricants. This has not been a problem with recorder bearings in past missions because the high-speed playback requirements have not been long enough to deplete the oil supply. However, with longer recording missions more in demand, this will be a problem if lubrication requirements are not accurately calculated and supplied to compensate for this loss.

Contaminated atmosphere and extreme temperatures will cause bearing failure if not controlled. These environments have not been a problem with spacecraft recorders for two reasons:

1) The spacecraft recorders are hermetically sealed;

2) They have been located in thermal environments that have been controlled within design operating specifications.

The hermetically sealed enclosure protects the bearings from external contamination, space vacuum outgassing, and lessens the lubrication evaporation process that is normally accelerated in a space vacuum.

Another problem noted during vibration-qualification testing is a condition known as bearing false brinelling or pitting. This is caused by the bearings being subjected to high vibration, in the presence of oxygen, while in a non-operational state. The solution is to operate the recorder while it is exposed to high vibrational environments.

An investigation of the bearing state-of-the-art by Ball Brothers Research Corporation is somewhat optimistic. In 1960, BBRC set out to achieve long bearing life in a vacuum to support the NASA Orbiting Solar Observatory Program. A theoretical model was developed by N. M. Fulk using the concept of quantum electrodynamic interaction to explain surface forces. This model was the basis for the long-life space vacuum lubrication process known as Vac-Kote. Vac-Kote is a technology that puts a dielectric between two surfaces to prevent destructive energy transfer. With this process, BBRC is now confident of achieving a five-year life in a vacuum, which is a more severe environment that encountered in a pressurized tape recorder. A continuous three-year vacuum life is predicted for a 2500 rpm application. BBRC is confident of a ten-year bearing life in a sealed tape recorder using some type of reservoir system.

Standard design provides lubricant reservoirs to replenish losses and the reservoir volume is designed to provide lubricant for at least 10 times longer than the system design life. One such reservoir system, attached to the bearing housing, is composed of a nylasint matrix impregnated with lubricant. The very low vapor pressures of the majority of space lubricants also regards lubrication depletion. Chapter II contains a discussion of lubrication systems.

The fluid-filled type of recorder immerses the tape, heads, motor, and bearings in lubricant, i.e., a totally enclosed tape transport chamber filled with a compatible lubricant. This eliminates tape/head contact and ensures a sufficient lubricant supply for bearings. The tape/head spacing is small, but signal reduction is experienced. With lower tape speeds, some sort of forced circulation system is required; it could be highly redundant to improve reliability. The major disadvantage is that the head-tape separation limits the packing density to about only 2000 bits per inch. Another possible disadvantage is the fluid is a fluorocarbon and may have some fluorine in it which is not compatible with polymer tape. However, life tests have demonstrated 50,000 successful tape passes. It is understood that lack of funds have caused cessation of most fluid filled transport development efforts.

Two companies claim some patents on fluid-filled designs. A fluid-filled recorder was used on Titan II. This type of design may be the solution to the 10-year life problem and should be investigated.

Air Bearings - Air bearings should be considered for tape recorders that operate continuously. Because of the bearing wear that occurs with many starts and stops, air bearings are not normally employed for intermittent operation. See Chapter IV on *Gyroscopes and Bearings* for a discussion on the application of air bearings. Since many aerospace magnetic tape recorders that are hermetically sealed use motors at reasonably high rpm and are pressurized with an inert atmosphere, air bearings would be feasible. The survey failed to disclose the current use of any air bearings in tape recorder applications. Loss of the hermetic seal would cause bearing failure. Also the effects of any lubricant in the tape could create problems. It is suggested that future tape recorder studies include determining the feasiblity of air bearings.

b. Survey Results - Two surveys were conducted. One survey involved general questions concerning the long life assurance of tape recorders; the consensus of the answers is presented in Table 4. The answers are incorporated in the discussions throughout the chapter. The second survey determined why specific selected recorders obtained long lives and high reliabilities. The results of this survey are summarized in Table 5.

None of the aerospace airborne recorders have been developed for really long life--five or more years. The OAO 11 operating life of three years was beyond expectations. The OGO 1 was in orbit over 5-1/2 years but recorded for only 3400 hours during this time. The longer lived satellite recorders have resulted from design evolution rather than a single long-life advancement effort. This is not to say analyses and design of long-life tape recorders has not been conducted. NASA/GSFC, JPL and IITRI, among others, have engaged in these types of efforts. There is general agreement concerning many areas of long-life design, test, and process control. These are presented in the guidelines for long-life assurance and discussed in this chapter.

Because of the small sample sizes and specific mission usage of aerospace magnetic tape recorders, it is difficult to draw statistical conclusions. The OAO 11 was chosen as one of the specific recorders because it functioned continuously in space longer than any other--six days short of three years. It incorporated many features not considered conducive to long life.

c. Hardware Life - One of the questions asked during the general survey (Table 4) was regarding the life of long-life spacecraft tape recorders. The consensus was that a two-year continuously operating spacecraft tape recorder is currently possible; the storage rate capability and bit error rate are within specification after two years continuous operation. Most stated that the state-of-the-art exists to build a low-speed record, medium-speed playback tape recorder that would operate continuously for five years. The life of higher speed recorders, naturally, will be proportionately less, because the higher the speed, the sooner the wearout point is reached on the life-limiting mechanical parts. Some believe that recorder life is so tied to the amount of tape passed that 10-year calendar life can only be obtained by long periods of inoperation. One source believes strongly that continuous operation of over 10 years is possible with a fluid-filled design. If for no other reason, this argument has merit in that those who disagree with fluid-filled designs believe five years is about the present limit.

Table 4 Results of Manufacturers/Users Survey Magnetic Tape Recorder

Questions	Consensus Answer
1. What are the major life limiting parts?	Heads, tapes, bearings, belts, clutches and brakes.
2. What is the life of available aerospace recorders?	Two years continuous operation in space. A five-year life, using slow speed record (1 ips) medium speed (15 ips) replay, is within the current state-of-the-art. (However, the recording characteristics of this "slow speed" machine may not be satisfactory for some missions.)
3. What are the expected head lives?	A function of transport design, head material, and feet of tape passed. Ferrite head life about 3 times that of aluminum, other parameters equal.
4. What are the wear characteristics?	Once a head starts to deteriorate from wear, it wears out rapidly. Debris on head builds up rapidly. Wear is a function of amount of tape passed, its abrasiveness, and interface pressure. Head geometry, such as wrap angle also affects wear (a small angle is desirable).
5. Is erasing the tape with permanent magnetics satisfactory?	Depends; may need an ac erasure that is programmed to allow more than one replay. Digital saturation recording does not require erasure. Keep away from ferrite heads that could become permanently magnetized.
6. Are there any essential failure mechanism differences between rotary and linear heads?	No; but because of its complexity the life and reliability of the rotary head is less than that of the conventional head.
7. Will the more sensitive tapes allow lower tape speeds?	Yes; but cannot use with all existing recorders. High energy tape is more difficult to erase and more sensitive to debris, can get a wider bandwidth. One company has obtained 46,000 bits/inch with the 971 series.

Table 4 (concl)

Questions	Consensus Answer
8. Would a tape wear-in, prior to use, have a beneficial effect such as reducing abrasiveness?	Yes; but opinion varied as to the number of wear-in cycles and the method of wear-in. Number of recommended cycles varied from 10 to 200. Some believed tape should be worn-in against a knife edge; others a rod or head.
9. Is Kapton superior to Mylar?	Kapton has superior tensile strength at elevated temperature than Mylar; but is more humidity sensitive than Mylar.
10. What problems are involved in using lubricated tapes?	Lubricant on oxide side should be limited to 2% to prevent binder breakdown and slippage (jitter).
11. What is the maximum instrumentation tape life?	About 50,000 passes are possible with a properly designed transport.
12. Should drive belts be avoided?	Yes; use direct drive.
13. Will fluid filled transports alleviate many long-life problems?	Yes; but recording density is low (1,200 bits/inch). Approach needs additional resources for development.

Table 5 Survey of Users and Manufactuers on Specific Parts

PART DESCRIPTION AND USING PROGRAM	USERS OPINION ON WHY PART WAS SUCCESSFUL	MANUFACTURER"S OPINION OF WHY PART WAS SUCCESSFUL	RECOMMENDED GUIDELINES IMPROVEMENTS, PROCUREMENT SPEC REQUIREMENTS, ETC.	RELATIVE IMPACT ON YIELD & COST OF INCORPORATING RECOMMENDATIONS
OAO 11	Configuration is an adaptation of recorder for OGO. This is a complex digital recorder that costs in excess of $125,000 each. Much of the cost due to the "tender loving care" given by manufacturer. This recorder configuration violated many current long-life guidelines—brass heads were used, concentric 1/2 inch reels used, and transport was complex. Clutches were eliminated by using a planetary arrangement driven by two separate motors.	This recorder functioned continuously about 25,000 hours on the orbiting astronomical observatory. It employed belts, gears, brass heads and small bearings which are not considered desirable for long life applications —a "contradiction" to its long life of almost three years. The manufacturer believed that because of the small sample size, delineation of what promoted success the most is not possible. However, the following "idealized" design rules were suggested: a. Minimize the number of moving parts; b. Avoid the use of high-speed shafts; c. Allow rubbing only at the head-tape interface; d. Avoid edge guidance of the magnetic tape; e. Contact recording surface of tape only at heads;	1. Replace brass head with alfesil or equiv. 2. Reduce complexity. 3. Continue close process control ("tender loving care") 4. Incorporate the idealized design rules delineated by manufacturer.	1. Costs about 1/3 more, but may get three times the use. 2. Could reduce costs. 3. May double the cost of the recorder. 4. Increased design costs could equal manufacturing savings on part elimination.

Table 5 (cont)

PART DESCRIPTION AND USING PROGRAM	USERS OPINION ON WHY PART WAS SUCCESSFUL	MANUFACTURER'S OPINION OF WHY PART WAS SUCCESSFUL	RECOMMENDED GUIDELINES IMPROVEMENTS, PROCUREMENT SPEC REQUIREMENTS, ETC.	RELATIVE IMPACT ON YIELD & COST OF INCORPORATING RECOMMENDATIONS
OAO 11 (Continued)		f. Avoid very small precision ball bearings; g. Don't use belts, clutches, brakes, gears, pressure pads, pinch rollers, brushes or slip rings; h. Provide adequate but not excessive lubricant and prevent it from migrating; i. Optimize gaseous environment; j. Use non-contacting optical end of tape sensing devices; k. Avoid the use of age sensitive material. ℓ. Keept cyclic stresses well within fatigue limits, and; m. Minimize the use of polymers which may have possible outgasing, curing, thermal or deterioration problems.		

Table 5 (cont)

PART DESCRIPTION AND USING PROGRAM	USERS OPINION OF WHY PART WAS SUCCESSFUL	MANUFACTURER'S OPINION OF WHY PART WAS SUCCESSFUL	RECOMMENDED GUIDELINES IMPROVEMENTS, PROCUREMENT SPEC REQUIREMENTS, ETC.	RELATIVE IMPACT ON YIELD & COST OF INCORPORATING RECOMMENDATIONS
HDRSS (Nimbus III)	Inspect belts under polarized light and at least 50 times magnification to see stresses and any edge damage.	Similar to OAO, but with improvements. Has completed about 12,000 hours of non-continuous operation. Nimbus recorders have accumulated over 8400 hours of operation time.	1. Inspect belts under polarized light and mangnification for damage.	1. Slight cost increase
			2. Eliminate end-of-tape mechanical switches, use optical means.	2. Small cost increase
	Used IITRI guidelines to reduce tape stress, flattened crowns.	Subjected to a strong life testing program. HDRSS passed 4229 REC/PB cycles accelerated life test and 4290 REC/PB real time life test. Used "tender loving care" in assembly. Assembled in class 100 clean room. Used hard faced head.	3. Pressurize transport with nitrogen or argon. Relative humidity range 15% to 30%.	3. Transports normally pressurized anyway --slight costs increase.
	Semi-conductors burned-in 300 hours.		4. Add a strip of abrasive tape to the end of the magnetic tape.	Slight cost increase
	Eliminated EOT switches, used optical devices.		5. Assembly in Class 100 clean room.	5. It is assumed initial capitol expenditure amortized over many programs, certified people are available and tools must be cleaned anyway. The labor costs for assembly and testing will be about 50% above that for non-clean room assembly and test.
	Development problems were: a. Tape sticking to the head cured by head selection and contour and by controlled humidity of 30% in an atmosphere of 78% N_2, 12% O_2 and 10% He pressurized to 17 psia. The tape becomes too smooth at lower humidity and too sticky at higher humdity.			

Table 5 (cont)

PART DESCRIPTION AND USING PROGRAM	USERS OPINION OF WHY PART WAS SUCCESSFUL	MANUFACTURER'S OPIONION OF WHY PART WAS SUCCESSFUL	RECOMMENDED GUIDELINES IMPROVEMENTS, PROCUREMENT SPEC REQUIREMENTS, ETC.	RELATIVE IMPACT ON YIELD & COST OF INCORPORATING RECOMMENDATIONS
HDRSS (Nimbus III) (cont)	b. Progressively reduced output caused by oxide buildup on the playback head. Cured by addition of a strip of abrasive chrome oxide "green tape" to the end of the magnetic tape so that the head is periodically cleaned. Enough time has not ensued to determine the effectiveness of this fix. c. There was so much variation in the quality of purchased tape that RCA made their own with adequate processing control. This included a maximum ambient temperature of 35°C. d. Erratic reproduction caused by tape pressure from the end-of-travel (EOT) switches. This was cured by using an optical EOT switch in conjunction with a transparent section of tape and backed up by a mechanical switch beyond the "green tape."			

Table 5 (cont)

PART DESCRIPTION AND USING PROGRAM	USERS OPINION ON WHY PART WAS SUCCESSFUL	MANUFACTURER'S OPINION OF WHY PART WAS SUCCESSFUL	RECOMMENDED GUIDELINES IMPROVEMENTS, PROCUREMENT SPEC REQUIREMENTS, ETC.	RELATIVE IMPACT ON YIELD & COST OF INCORPORATING RECOMMENDATIONS
HDRSS (Nimbus III) (cont)	e. Broken drive belts cured by rigorous inspection. The standard construction of these belts for recorders is a flat doughnut shaped ring punched from 0.0005 inch mylar and turned to fit on pulleys. f. Flutter caused by resonance of the mechanical drive including 5 drive belts, a planetary drive, two motors, and ball bearings. Reduced but not cured by mechanical de-tuning and by replacing, readjusting, and cleaning elements. g. Pressure leak at connector O-ring seal due to lake of key permitting distortion of ring when connector lock nut was tightened. Eliminated with proper parts.			

Table 5 (cont)

PART DESCRIPTION AND USING PROGRAM	USERS OPINION ON WHY PART WAS SUCCESSFUL	MANUFACTURER'S OPINION OF WHY PART WAS SUCCESSFUL	RECOMMENDED GUIDELINES IMPROVEMENTS, PROCUREMENT SPEC REQUIREMENTS, ETC.	RELATIVE IMPACT ON YIELD & COST OF INCORPORATING RECOMMENDATIONS
HDRSS (Nimbus III) (cont)	*Flight failures* were gradual deterioration of outputs from same three channels on both recorders sup-positionally caused by oxide buildup on part heads in rela-tively light con-tact with the tape due to its canting by the mechanical drive configuration. This should be cured by use of the "green tape."		6. Use NASA/GSFC specification S-715-P-14 as a guideline for selecting and using magnetic tape.	6. Over 2/3 of the tape tested will be rejected dur-ing acceptance testing. A reel costs up to $50. The tests are de-signed to be con-ducted by tech-nicians. The test costs will be relatively small;however, if more elaborate tests are deemed necessary to demonstrate quality for longer life missions, costs will rise. S-715-P-14 tests are scoped to ob-tain tapes good for 10,000 cycles.

Table 5 (cont)

PART DESCRIPTION AND USING PROGRAM	USERS OPINION ON WHY PART WAS SUCCESSFUL	MANUFACTURER'S OPINION OF WHY PART WAS SUCCESSFUL	RECOMMENDED GUIDELINES IMPROVEMENTS, PROCUREMENT SPEC REQUIREMENTS, ETC.	RELATIVE IMPACT ON YIELD & COST OF INCORPORATING RECOMMENDATIONS
Leach Model 2351 (Military Satellite)	Hard faced head gave improved life. Proper lubricant employed. Used some of the long life guidelines mentioned for other recorders.	Successfully completed 16,500 hour life test and two years in orbit at an 85% duty cycle. During life tests at Goddard, had a bearing failure after 5000 hours due to lubricant depletion; replaced with ANDOC C grease that performed okay. Life of recorder obtained thru evolutionary development of other recorder designs. Used hysteresis AC motor; believed servo controlled DC motor may be unacceptable except for digital recorders. They burn-in electronics 100 to 200 hours. Transport worn-in about fifty hours to eliminate infant mortaility. Used hard faced heads. Tape wrap angle on capstan drive should be about 270° unless capstan has an unusually high coefficient of friction rubber is not good since it gets lumpy	1. Burn in electronics 100-200 hrs. 2. Wear in transport. Two hundred passes suggested, requiring over fifty hours operation if tape speed is low.	1. Normal procedure. No impact. 2. Small cost increment--such as two man weeks labor.

Table 5 (concl)

PART DESCRIPTION AND USING PROGRAM	USERS OPINION OF WHY PART WAS SUCCESSFUL	MANUFACTURER'S OPINION OF WHY PART WAS SUCCESSFUL	RECOMMENDED GUIDELINES IMPROVEMENTS, PROCUREMENT SPEC REQUIREMENTS, ETC.	RELATIVE IMPACT ON YIELD & COST OF INCORPORATING RECOMMENDATIONS
Leach Model 2351 (Military Satellite) (cont)		and sticky. No large and complex recorders have been built for unattended operation--Skylab recorders need maintenance.		

Slow-speed recording is defined as operating at 1 ips or less; medium-speed recording is operating between 1 and 15 ips; and high-speed recording is above 15 ips. IITRI does not recommend operational tape speeds below 1.0 ips because a tendency to stick-slip at these low speeds. Special missions that used low speed (Mariner '71) were not long-life missions. (Ref. G. S. L. Benn: *Mariner Mars Head/Tape Stick Slip Study*, IITRI Project E6169, February 1971.)

The survey revealed that specifications on size, weight, power, flutter, tape speed, etc., have usually forced design approaches that have not optimized recorder life. Magnetic tape is considered to be the pacing life-limiting component in the recorder by most recorder manufacturers. For manned missions, the magnetic tape would not be the life-limiting component if tape replacement is allowed. The magnetic tape in a properly designed transport probably will not need replacement before five years; the earliest allowable maintenance point permitted by the ground rules of this study. No data were found on tape calendar life excluding head wear. Therefore, it is assumed the tape has a life of at least 10 years excluding tape/head interface wear.

Figure 1 identifies the tested versus the predicted life (with state-of-the-art improvement) of the life limiting parts in the spacecraft tape recorder. The shaded areas in Figure 1 are based on actual recorder and recorder component test data. The unshaded areas are the predicted life extension that should be expected with current technological advancements.

The OAO II tape recorder which functioned from mid 1968 until December, 1971, has had about 25,000 hours of in-orbit continuous operation. The OGO 1 (EGO) recorder has accumulated over 3400 hours of operation during its 5-1/2 years in space (Reference 7). A number of recorders have operated over five years in space. The data indicates that building a tape recorder capable of operating in excess of five years possible, although long-life is not achieved with regularity.

As indicated in a previous section, REL logged more than 25,000 tape cycles on a spacecraft recorder during laboratory long-life testing. The general consensus of the general survey was that technology exists to obtain 50,000 tape cycles with improved tapes and transport design.

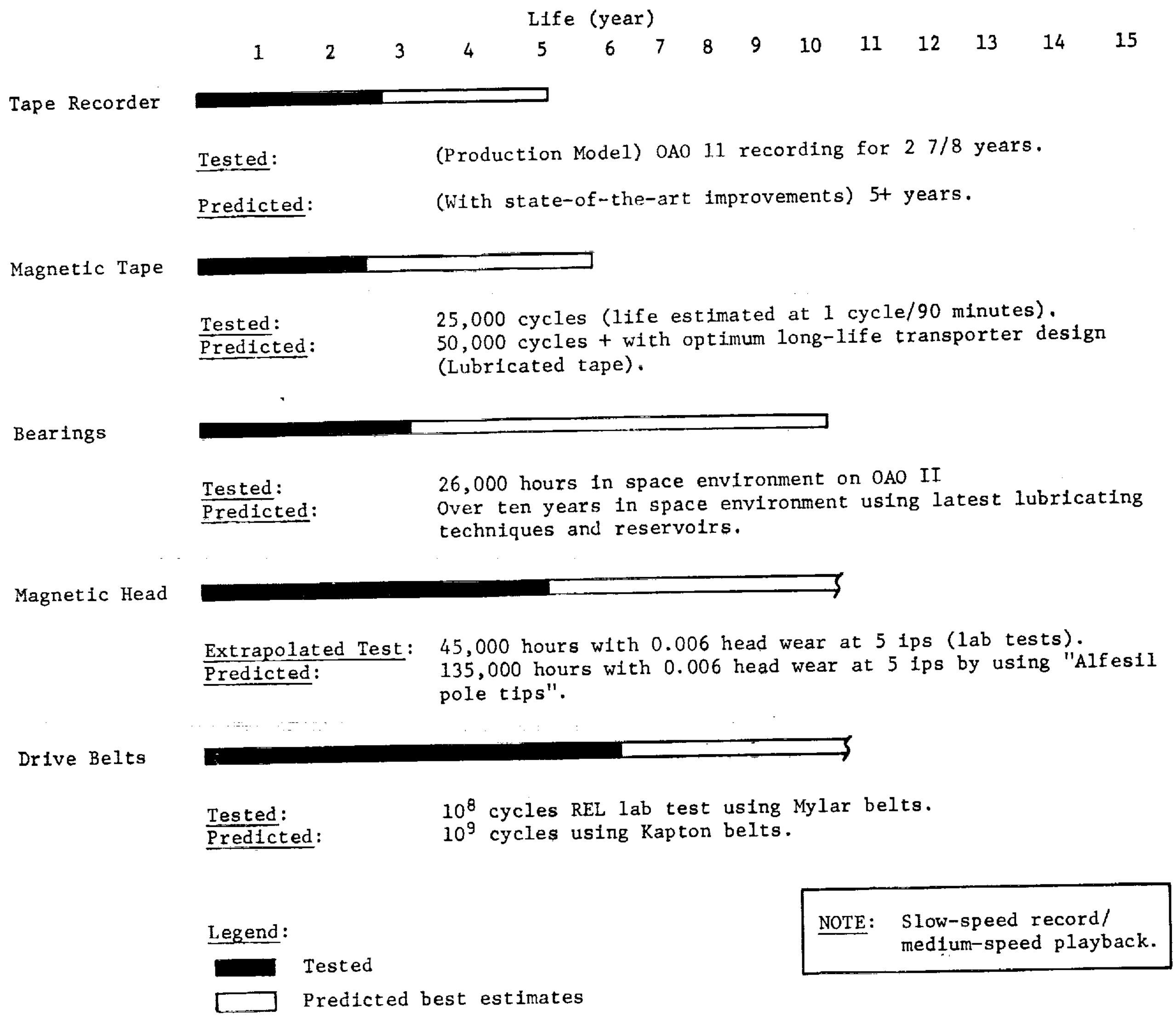

Figure 1 Tested vs Predicted Life of Spacecraft Tape Recorders and Parts

The tape recorder bearings in the OAO-II operated three years before failing due to depletion (assumed) of lubrication. A bearing life in excess of ten years is predicted for spacecraft recorders using some of the latest lubricating techniques such as film barriers, self-lubricating ball retainer, reservoirs, etc.

Data for the magnetic head (Figure 1) were obtained from a West Coast magnetic head manufacturer.

Drive belt data was derived from testing at REL. Data for the unshaded areas were based on extrapolated test data and theoretical analysis. Belt life is a function of the fatigue and can be predicted on the basis of combined stresses. Because belts are stressed by bending around pulleys and in transmitting torque, fatigue limits can be determined by combining the bending and the tensional stresses. The belt material will retain its initial tension if it is not stressed sufficiently to cause creep.

In a JPL/Kinelogic test program, relationship between life and stress ratio was developed. Stress ratio is defined as the maximum stress, less the minimum stress, divided by the endurance limit of the belt material. According to these life tests, the belts must be operated at stress ratios of less than 0.86 to achieve 10^9 cycles.

Electronic parts were not considered to be life-limiting problems if they were maintained at a temperature below 50°C. Outgassing of some epoxies occurred above 50°C during testing at NASA/Goddard. If the temperatures cannot be controlled, this problem can be solved by separating the heat producing electronic equipment from the transport.

With the exception of enclosure seals, the long-life problems identified parallels the findings of a NASA/Goddard committee created to study spacecraft tape recorder failure/problem history. This history was compiled from tests of 75 tape recorders used on the Tiros, Nimbus, Mariner, and OGO satellite programs. The percentages of the above mentioned problem areas relative to the total number of failures experienced during the said testing are tabulated.

Mechanical	Percent
Enclosure Seal	20
Magnetic Tape	15
Magnetic Head	12
Bearings	8
Drive Belts	7
Other	13
TOTAL	75
Electrical	25%

No repetitive failure modes were experienced in the electrical system with the exception of those mentioned above.

The high percentage of enclosure seal failures resulted from quality control problems experienced by one recorder manufacturer and are not considered to be a pacing constraint with long-life spacecraft tape recorders.

d. *Application Guidelines and Alternative Approaches* - In the final analysis, the final design of any component must be selected from among competitive alternatives via trade-off studies, knowing specific mission objectives and constraints. The magnetic tape recorder is no exception.

For unmanned missions, in-space maintenance and restoration is not normally possible. Standby redundancy (recorder or buffer) can be employed if weight constraints permit. Current technology can produce a recorder for a five-year space mission. For longer space missions it may be necessary to either go to a recorder with a fluid-filled transport, or employ redundancy.

More options are open to the manned space missions, especially a program such as Shuttle where land based maintenance and restoration is periodically available. Some of the options are:

1) Periodically replace a life limited off-the-shelf aerospace recorder and save development costs of a long-life recorder.

2) Expend resources and develop a long-life recorder and save on maintenance time.

3) Use standby recorders, or buffer, at the expense of weight and complexity to backup off-the shelf aerospace recorders.

4) Replace the magnetic tape periodically and other life limited components, as required. One approach is the "Utility Modular Maintenance" concept where the tape, head, bearings and other life limited components are periodically replaced as a modular unit as required.

5) Develop and use a hi-rel, long-life recorder to record critical functions that must be telemetered (after time compression). Use an off-the-shelf recorder for non-critical engineering data that will not be telemetered but retrieved after the spacecraft returns to earth.

6) Build the data system around the tape recorder, recognizing the tape recorder is the weakest link. (A suggestion by RCA's Mr. Ligon.)

JPL is pursuing two ideas for a data storage system which so far appear to be feasible and workable for some applications:

1) Eliminate multispeed requirements by using a semiconductor or magnetic buffer for input/output of data and parallel recording on multiple tracks.

2) Simplify transport further by using three electronically regulated dc motors to maintain low and constant tape tension. This requires only three low-speed 200 rpm's rotating assemblies with no belts.

Data can be stored and retrieved by means other than magnetic tape recorders. The challenge to magnetic recording will come from research on static mass data storage techniques. It remains to be seen whether advances in this field will eventually displace the tape recorder from its present dominant position. It is suggested that a study be conducted to determine the adequacy of these other means for spacecraft applications. Trade-offs between these means should be made. In additon to laser recording techniques previously mentioned (optional/thermal mass memory), magnetic discs, silicon drum, dynamic MOS RAM's, domain wall technology, magnetic bubble technology and other data storage/retrieval methods should be considered in a parametric analysis. Parameters to be considered (besides mission constraints, life and reliability) include bits per dollar, access rate, weight, value and maintainability. Also, the maturity of the alternative to the tape recorder must be considered; an alternative only in the laboratory phase should not be considered for present use because of the low probability of success.

The key to high storage density is the submicron wavelength of light because the theoretical limit on the closeness of stored bits to each other is set by the wavelength of the recording energy. Conventional magnetic recording is limited to relatively long wavelengths. The length is established by the size of the gap in the recording head.

Electro-optical technology will permit greater storage densities than magnetic tape. A trillion-bit memory the size of a golf ball is on the way with electro-optics. Storage densities of 10^{12} bits per square centimeter have been achieved at Radiation, Inc. RCA has achieved similar densities with holograms using bismuth and arsenic trisulfide.

Laser and electron beam digital recording techniques are being developed. A laser beam is focused upon a metalized tape and vaporizes a spot. To read the recorded information, the power of the laser is reduced at the point where there is a bit and, at this reduced level, the light is reflected back. Optical methodology is available to check that a data bit is actually recorded. The Precision Instruments laser mass memory system can store 10^{12} bits of data on a 4.75 inch by 31.25 inch long tape.

Most tape manufacturers include silicon or other material in their formulation for lubricating the tape-head interface. Other lubricant possibilities that should be investigated in future work is gaseous lubricants such as CO_2 and SF_6 and total immersion of transport in a low viscosity silicone fluid.

Mr. Klechefski, 3M Company, suggested that tape be stored at 30% RH and at a temperature less than 32.2°C. Bring the storage environment up to the service environment over a 24 hour period prior to use. He also suggested that the tape be stored in an argon atmosphere to further retard aging degradation. Drive belts can shrink in storage, and therefore, should be inspected prior to usage.

D. TEST METHODOLOGY AND REQUIREMENTS

Since the major failures modes involve mechanical wear, qualification tests should emphasize mechanical life verification. If the recorder service operation is intermittent, the wear mechanisms can be accelerated by operating the recorder continuously at its highest normal (replay) tape speed. Increased tape tension can also accelerate wear. However, operating at tape velocities and tensions well above that normally employed by the recorder can provide erroneous wear results. The high tape tensions (say 2000 psi) can generate temperatures and wear disproportionate to the increased tension. For example, ferrite heads, with their poor thermal conductivity, could wear disproportionately faster at high tape tensions. On the other hand, some wear mechanisms are more severe at low rubbing speeds because at higher velocities, interfaces may become separated by aerodynamic forces and thermal dissipation improved from increased mass movement. The foregoing is true of the head-tape interface and, within limits, bearings. The fluid filled transport reduces wear by introducing a thin film of inert fluid between the tape and the head.

For accelerated wear tests, it is suggested that the normal operating temperature of the head and bearings be determined. The tape velocity is increased until a head or bearing temperature is obtained that is slightly below that where the material hardness (wear) characteristics rapidly degrade. Operate at this velocity until wearout is obtained, providing a wearout life indication in total number of tape feet past the head or the total number of bearing revolutions. Tape tension can be also increased to accelerate head wear, but the correlation factor between real-time and accelerated test conditions will be more difficult to access than velocity-wear relationships.

If the magnetic tape recorder operation during a mission is not continuous, the Dynamic Mission Equivalent (DME) may be used to reduce test time per mission simulation cycle. In general, the DME approach is:

1) Define mission profile;

2) Subdivide profile into dynamic (non-equilibrium) and static (equilibrium) portions;

3) Eliminate/reduce static portions where all parameters are simultaneously quiescent;

4) Compress the time profile when one or more parameters are always dynamic (changing) by accelerating the rate of change in the parameter (but only when component response is not affected);

5) Re-define the mission profile for test purposes incorporating items 3) and 4) above.

Reference 8 presents an example of DME application to a Grand Tour type mission where the test time is reduced by a factor of 663 to 1. This example is part of an accelerated testing study. Volume IV of this study (NAS9-12359) presents the results of an analysis of accelerated testing methodology applicable to tape recorders among other components.

The magnetic tape lends itself to screening and wear-in techniques. The tape can be screened for acceptable mechanical and electrical properties using standard techniques. Tape tensile strength and elongation under load and elevated temperature (50°C maximum for mylar) can be checked for compliance. Following the mechanical tests, the tapes can be checked for electrical compliance for distortion and bit dropout rate, etc. Using rigid specifications for selecting quality tape, two recorder manufacturers rejected 70% of the tapes.

As described previously, tape should be worn in to reduce abrasiveness. Opinions vary as to the number of cycles and the means of wear-in. One person believed that the wear-in obtained during Q&A tests was adequate. Two persons believed ten passes were sufficient, some 200 passes. Some burnish the tape against a knife edge, while others wear-in the tape against the head with which the tape will see service. Some believed that wear-in also reduces data drop-out.

Martin Marietta Aerospace screened 250 magnetic tape reels after a wear-in against a magnetic head. The wear-in reduced error. Stability was usually obtained after 5 to 10 passes. They attempted wear-in by dry wipe, ceramic head, burnishing, knife edges--all these approaches increased the error rate. The only improvement came from running the tape against its head.

In view of the diversity of opinion, it is believed that the more conservative NASA/GFSC guideline in S-715-P-14 should be followed. A break-in period of 200 passes is advisable. The break-in should be with the head it will be used with. Observation of a deep scratch, indicating repeated damage by debris in the same area should be cause for replacing the tape and relapping the head. The head should meet the surface finish requirements at the conclusion of the break-in.

The tape should be tested for thermal stability prior to acceptance. Remove a ten inch sample length of tape from subject reel. Avoid contaminating the oxide surfaces. Fold sample in half, oxide surface to oxide surface. Rub test sample against a 3/8 inch diameter rod heated to 175°C for five seconds. There should be no evidence of adhesion or oxide crumbing. See GSFC Specification S-715-P-14 for details of test and acceptance criteria.

JPL utilizes a durability test. THF is used to clean off the oxide. The adhesion is considered good if the oxide remains on 25-27 seconds, poor if the oxide remains 10 seconds or less. The test is operator sensitive.

E. PROCESS CONTROL REQUIREMENTS

The consensus of opinion was that cleanliness in assembly is essential. Class 100 clean rooms are suggested. Store parts in nitrogen filled polyethline bags. One comment was that even a puff of cigarette smoke can produce data dropouts.

It is important that the gap and appex of the head closely align. These can be aligned by observing and maximizing the amplitude of the signal. Misalignment is more critical with the new harder alloy heads since misalignment cannot be "lapped-in" from passage of the tape like the softer heads could.

Many of the processes involved in the manufacture of magnetic tapes is proprietary. Hence in many cases, it is best to specify and test for required characteristics such as roughness (1 - 3 microinches), lubricant, flash temperatures, etc, rather than attempt to control processes. Use Reference 6 as a baseline. Guard against the formation of nodules during manufacturing. Some users require the reel to be heated (baked at 40°C) prior to usage to obtain additional shrinkage and dimensional stability; because of a phase change, the tape shrinks slightly upon heating.

Use of inadequate quality control selection techniques is one cause of bearing failure. Most of the spacecraft tape recorder industry has adopted rigid inspection techniques and procedures for bearing selection because the bearing manufacturers have no inspection techniques suitable for space applications.

The importance of rigid inspection techniques can be illustrated by Ball Brothers Research Corporation (BBRC) experience. BBRC adopted a rigid inspection technique for selecting flight hardware. Rejected components with insignificant flaws are used in ground equipment that does not have stringent operating requirements. Consequently, four failures have occurred in ground equipment and no failures have occurred in flight hardware which has had many more operating hours than the ground equipment.

According to BBRC, long-life verification of the treated bearings is not necessary or practical. Even if several bearings completed life testing, the small sample size could be statistically questioned. BBRC's approach to long-life verification is to:

1) Select the proper bearing for the design using the traditional standards developed by the Anti-Friction Bearing Manufacturer's Association;

2) Select a stable lubricant with a 5-year (or longer) life and prove this by accelerated chemical tests and analyses;

3) Assure that the lubricant will remain on the interface by short-duration tests and analysis;

4) Conduct an accelerated life test on a prototype model to prove compatibility of lubricant, bearing, and application;

5) In production, employ screening tests and inspections to eliminate marginal bearings as received by the bearing manufacturer.

F. REFERENCES, BIBLIOGRAPHY AND CONTACTS

1. References

1. M. A. Perry: "Magnetic Recording - Latest Developments in Ground and Airborne Systems." *Electronic Engineering*, September, 1970, p 48.

2. "Who Has Problems? Technical Analyses: Recorders." *Telemetry Journal*, December/January, 1971, p 28.

3. J. R. Hays and D. Knell: "Wideband Image Recorder for ERTS." *International Telemetry Conference Proceedings for 1970.*

4. G. S. L. Benn and Dr. R. L. Eshleman: *Design Study of a High Reliability Five Year Spacecraft Tape Transport* IITRI, Chicago, Illinois, E 6179, November, 1971.

5. J. F. Carroll, Jr. and R. C. Gotham: "The Measurement of Abrasiveness of Magnetic Tape." *IEEE Transactions on Magnetics, Vol 2, No. 1*, March 1966, pp 179-183.

6. Carl Powell: *Specification for Selecting and Using Magnetic Tapes and Heads in Satellite Tape Recorders.* NASA/GSFC S-715-P-14, February, 1972.

7. R. Miller, R. T. Treadwell: "Long-Life Spacecraft Tape Recorders." *IEEE Electronics and Aerospace Systems Convention*, Washington, D. C., October 26-28, 1970, pp 107-112.

8. *Handbook of Criteria for Obtaining Long-Life Space Vehicles.* T-71-48891-004. Martin Marietta Aerospace, Denver, Colorado, December 31, 1971.

2. Bibliography

C. R. Hume, R. Miller, R. J. Treadwell: "Long-Life Spacecraft Tape Recorders," *British Interplanetary Society Symposium on Application Satellites*, April 14, 15, 1971.

R. Shatavsky: *Design, Development, Fabrication and Test; Improved Magnetic Tape Recorder.* JPL, Pasadena, 66/10/31, California, ASA-CR-92550 (JPL-951128).

I. Krash: *Comparison Fatigue Life of Seamless Polyester/Film Belts Fabricated by an Alternate Source.* JPL, Pasadena, California, 66/11/10, ASA-CR-92519 (JPL-377029).

Benn D. Martin: "Data Subsystems for 12-Year Missions." *Astronautics and Aeronautics*, September, 1970.

John D. Hernzmann: "Spacecraft Tape Recorders." *Instruments and Control Systems*, February, 1966, p 92.

L. D. Berringer: "Aerospace Tape Recorders." *Instruments and Control Systems*, April, 1966, p 97.

G. L. Davies: *Magnetic Tape Instrumentation.* McGraw-Hill Co., 1961.

K. W. Stark and A. F. White, Jr: "Continuous Loop Recorders." *Instruments and Control System*, April, 1966, p 101.

R. E. Haday: "Iso-Elastic Tape Drive." *Instruments and Control Systems*, April 1966, p 111.

J. D. Rittenhouse: "Rotary Head Instrumentation Recorders for Telemetry Recording." *Telemetry Journal*, October/November, 1968.

Marilyn Farley: "Pushing the Limits, Technical Analysis: Recorder." *Telemetry Journal*, October/November, 1971, p 11.

F. Jorgensen: "Investigation and Measurement of Magnetic Head/Type Interface Problems." *Telemetry Journal*, February/March 1972, p 13.

F. D. Kell: "Video Recorder/Reproducer for Space Exploration." *1968 IEEE National Telemetering Conference*, p 187.

"Review of Instrumentation Tape Recorders." *Instruments and Control Systems*, July 1970, p 67.

S. Broadbent: *Electric Motor Commonality and Trade-Off Study.* Martin Marietta Aerospace, TN-3770129, October 1971.

Survey, Electric Motors Used in Space. , Martin Marietta Aerospace, T-71-48890-004, December, 1971.

Devine, Evans & Leasure: *Instrument Bearing Performance in Space and Laboratory Vacuum.* NASA TN D-6035, October, 1970.

S. W. Athey: *Magnetic Tape Recording*. NASA SP-5038.

R. J. Owen: *Magnetic/Tape Interface Study for Satellite Tape Recorders*. IITRI, Chicago, Illinois, E6134, February, 1971.

Carl Powell: *Feasibility Model for a High Reliability, Five Year Tape Recorder for Unattended Earth Observation Satellites*. NASA/GSFC, S-715-P-6, March 4, 1971.

Other IIT Research Reports of interest are:

E-6205 - *Five Year Magnetic Tape for Unattended Satellite Tape Recorders.*

E-8116 - *A Satellite Recorder System - Engineering Services for Life Assurance.*

E-8115 - *Review of HEAO Satellite Recorder Procurement.*

E-9572 - *Analysis of Wear on Magnetic Heads.*

E-6188 - *Atmosphere Explorer Tape Recorder Design Evaluation.*

E-1155 - *Detection and Analysis of Satellite Tape Recorder Degradation.*

E-6199 - *MM'71 Magnetic Head/Tape Stick Slip Study.*

E-6004 - *Study of Factors Affecting the Life of Magnetic Recording Tape.*

Steve Thompson: "Memories, Bulk Storage." *The Electronic Engineer*, June 1971, pp 49-62.

Wallace B. Riley: "How to Minaturize Mass Memories." *Electronics*, February 14, 1972, pp 91-92.

Stephen V. Thompson: "Charting the Memory Market." *The Electronic Engineer*, January 1972, pp 29-37.

Wideband Instrumentation Magnetic Tape. Mil-F-8986, 1 February 1970.

Tape Recording, Instrumentation Magnetic Oxide Coating. W-T-001553 (GSA-FSS), March 5, 1971.

3. <u>Persons Contacted</u>

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VII. PLUMBING COMPONENTS AND TUBING

by P. J. Powell

VII. PLUMBING COMPONENTS AND TUBING

A. INTRODUCTION

This chapter examines the long-life characteristics of tubing and fittings. The various tube fittings and sealing techniques available for high pressure fluid systems are summarized. This study investigated the activities of aircraft, aerospace, airlines and the SAE A-6 Committee to identify the new permanently installed type tube fittings, as well as the presently used and anticipated tubing materials. The new qualified fittings include many welded and swaged joining methods. Qualification to MIL-F-18280 has been met by both permanent and reconnectable tubes and fittings.

This chapter covers permanent attached tube fittings which comprise three joining methods for attaching the fitting sleeve to the tubing to form a permanent bond. These methods are: (1) multicavity fluxless induction *brazing* (can be used on 21-6-9 tubing), (2) *welding* (can be used on 21-6-9, AM 350 and 321 stainless steel), and (3) *swaging* which can be used with several metallic tube materials.

Separable attached tube fittings are reconnectable and include AN flared, MS flareless, and various similar designs.

Most of the tubing material and joining techniques advancement has been developed for aircraft high pressure hydraulic systems. However, the adaptation to spacecraft, such as Shuttle, is apparent. The concerns of industry, at the present time, in addition to the standardization of materials and joining methods, are the agreement of a common test and design criteria.

B. GUIDELINES FOR LONG-LIFE ASSURANCE

The service lives of lines and fittings are good for ten years. Although spacecraft systems have not met this goal, aircraft systems have satisfied this requirement.

The life limiting failure modes for lines and fittings (listed in the order of most probable occurrence) are:

1) Leakage of separable or permanent joints;

2) Leakage of plumbing or hose, and;

3) Burst/rupture of separable joints, permanent joints, plumbing, or hose.

1. Design Guidelines

1) Maximize the use of permanent type joints. Improper installation techniques were noted as the major cause for tubing and fitting service failures. Permanent fittings:

    a) Represent a reduced weight when compared to separable fittings;

    b) Reduce system leakage;

    c) Automatic programming equipment produces consistent results;

    d) In case of an unacceptable brazed joint, the number of reheat cycles shall be limited to three;

    e) In case of an unacceptable welded joint, the number of reheat cycles shall be limited to one, and;

    f) A list of the advantages of both separable and permanent fittings are presented in Tables 5 and 6 in the test.

2) Suggestions for separable fittings;

    a) Seal problems can be solved by replacing the seal, if separate seals are employed,

b) Thread failures can be remedied by:

- Providing for nut replacement capability;
- Assuring the failure occurs in the replaceable nut. The threaded female flange must be made of harder material, and;
- Using stub ACME threads, they would reduce thread failures.

3) Design must require alignment fixtures for tubing. Otherwise proper alignment of parts cannot be made with the welding head or the braze fitting, during manufacture when joining tube ends.

4) Locate plumbing fittings and components so that the joints can be made in a horizontal axis for installation ease.

5) To avoid elastomer damage within components or fittings, the temperature gradient of tubing during brazing or welding operations must be a consideration.

6) Weld fittings shall be machined from plate stock only. Utilizing plate rather than bar stock will minimize the possibility of leakage.

7) Weld fluid associated components to next adjoining part rather than braze. This avoids final acid treatment on tube stubs after the individual component has been assembled.

8) Limit the use of hose. The use of flexible tubing reduces the system leakage.

2. <u>Process Control Guidelines</u>

1) Specify tubing ovality to be 3% maximum. Excessive tube ovality creates stresses which reduce the service life of the system tubing.

2) Cleaning process should require that tubes be rinsed with demineralized water and forced dried. Recent problems on the Skylab program indicated that solvent vapor degreasing resulted in stains and pitting on stainless steel tubing.

3) Brazing requires cleanliness such as a Code 3 clean area to affect a good joint.

3. Test Guidelines

1) It is recommended that testing techniques be developed which closely simulate actual usage conditions.

2) Failures generated in burst, impulse, or flexure rarely occur in service.

3) The major portion of field failures are related to: (See text for suggested solutions.)

   a) Tubing/fitting installation;

   b) Tubing/fitting fabrication, and;

   c) Long lengths of unsupported tubing that fail during vibration.

4) Advanced flexure testing methods developed for tubing/fitting problems are: (See test for details of methods.)

   a) Rotary or planar centilever flexure, and;

   b) Planar vibration free-free beam.

4. Special Considerations

1) Develop qualified in-flight leak repair technology for space applications.

2) The total length of piping used on the DC-10 is 53% more than that used on the DC-8. Important trends are evident:

   a) 70% of the B-nuts have been eliminated;

   b) Brazed or swaged joints are used instead of B-nuts by a factor of 4 to 1;

   c) Use of flexible tubing has increased;

   d) Use of flexible hose has decreased, and;

   e) 97% of "o" ring seals have been eliminated by the use of metal seals.

3) The commercial aircraft panel reported that the five major problems causing 52% of the aircraft delay were:

   a) "B" nuts;

   b) Seal replacements;

   c) Fittings;

   d) Flexible hose, and;

   e) Tubing.

C. LIFE LIMITING PROBLEMS AND SOLUTIONS

1. Failure Mechanism Analysis

Table 1 presents the Failure Mechanism Analysis (FMA) which lists the failure modes, mechanisms and solutions for plumbing fittings and tubing. The long-life problems that result from the failure modes and their potential solutions are discussed in the following paragraphs. The three failure modes listed in order of their most probable occurrence are:

1) Leakage (slow loss of media) from both separable or permanent joints;

2) Leakage (slow loss of media) from both plumbing or hose, and;

3) Burst/rupture (rapid loss of media) from separable or permanent joints, or plumbing or hose.

a. Leakage - There has been a tendency by some aircraft/aerospace builders to change from flared or flareless sleeve designs in an attempt to avoid some of their shortcomings which are:

1) Seepage/leakage between the sleeve and tube;

2) Torque sensitivity;

3) Excessive insertion length, and;

4) Inability to withstand repeated reconnections.

The airlines and military agencies have been continually concerned about leakage problems that occured with MS fittings. Demand for improvement motivated the Air Force and NASA to conduct development programs for new fittings. Meanwhile, various changes to MS fittings have been introduced as shown in Table 2 (from Reference 1 data). It presents the current aerospace and aircraft tubing and fitting applications.

The two most common failures of a separable coupling or fitting are a failure of the seal or threads, which results in excessive leakage. The seal problem can be remedied by replacing the seal if separate seals are employed. The thread failure, due to stripping or galling, can be a very serious problem if a backout solution is not available. A coupling configuration that provides a

Table 1. Failure Mechanism Analysis - Plumbing Fittings and Tubing

Part and Function	Failure Mode	Rel. Rank	Failure Mechanism	Detection Method	How to Eliminate/ Minimize Failure Mode
Separable joint (contains flow media)	Leakage/Seepage (slow loss of media)	1	Incomplete seal between the sleeve tube, or B nut. Thread failure due to stripping or galling.	Inspection/detection methods include: 1) visual 2) x-ray 3) gage 4) system tests prior to flight 5) system measurements during flight	1) Limit the number of times that a joint may be opened or closed. 2) Provide a positive stop for the fitting to eliminate the need for a close torque requirement. 3) Provide positive attachment of the tubing and sleeve. 4) Insure precise alignment of the fitting halves. 5) Threaded female flange made of harder material than the nut. 6) Provide spare nuts and a nut removal capability.
Permanent joint (contains flow media)	Leakage/Seepage (slow loss of media)	1	Incomplete seal between the sleeve and tubing or end butts	Inpsection/detection methods include: 1) visual 2) gage 3) x-ray 4) dye check 5) ultrasonic 6) system tests prior to flight 7) system measurements during flight	1) Limit weld pass-over to two times. 2) Strict cleanliness requirements are necessary for brazing. 3) The heat effected zone extends beyond the fitting envelope thus reducing the tubing strength. 4) Insure precise alignment of the tubing halves.

Table 1 (cont)

Part and Function	Failure Mode	Rel. Rank	Failure Mechanism	Detection Method	How to Eliminate Minimize Failure Mode
Plumbing and hose (transport flow media)	Leakage/Seepage (slow loss of media)	2	Permeability of hoses and tubing (cracks and pin holes)	Inspection/detection methods include: 1) visual 2) x-ray 3) system tests prior to flight 4) system measurements during flight	1) Use hose bonding techniques which preclude hose layer separation. 2) Bend radius limitations. 3) Provide a leak repair kit.
Separable and permanent joints, plumbing, and hose (contain and transport flow media)	Burst/rupture (rapid loss of flow media)	3	Pressure spikes, flexure, fatigue	Inspection/detection methods include: 1) visual 2) x-ray 3) system tests prior to flight 4) system measurements during flight	1) Require positioning of tubing and mechanical joint in place without any stress. 2) Minimize use of flex hose. 3) Provide system overpressure protection. 4) Provide support for fittings and tubing to control the affects of vibration and flexure.

Table 2 Currently Used Aerospace and Aircraft Tubing and Fittings

Vehicle	Pressure Line	Return Line	Fitting
B52 KC 135	304 Ann.	6061 T6	An Flared
707 727 737	304 1/8 h	6061 T6	MS Flareless
A4	304 1/8 h	" "	MS Brazed on Sleeve
A7	"	" "	MS Permanent: Brazed
747	21-6-9	" "	MS Swaged-On Sleeve Permanent: Welded
C5	AM 350	" "	MS Swaged-On Sleeve Permanent: Welded
Late Model 737	21-6-9	" "	Reduced Weight Unions and Nuts
DC 10	"	N.A.	MS Brazed-On Sleeve Permanent: Brazed, Swaged
L-1011	"	6061 T6	Reconnectable: Dynatube Permanent: Welds
F-14	3AL-2½V	3AL-2½V	Reconnectable: Same as L-1011 Permanent: Shrink Fittings in Sizes through-8 Brazed -10 and Over
F-15	"	6061 T6	Reconnectable: Same as L-1011 Permanent: Swaged
Titan	21-6-9 304 L	21-6-9 304 L	MS Flareless Permanent: Welded
Viking Orbiter	304 L	304 L	Brazed
Viking Lander	321	321	5-8% Reconnectable: Flared (P) 12-15% Brazed 80% Welded
Skylab	347 304 304 L	347 304 304 L	MS Precision Permanent: Brazed
LM	304	304	Gamah Fittings Permanent: Brazed

method for nut replacement in case of thread failure has a major advantage over the conventional couplings. The design includes a replaceable nut and split retainer ring for disassembly purposes. To be assured that the thread failure occurs in the replaceable nut, the threaded female flange would be made of harder material. To remove a faulty nut, the nut would be moved back over the male flange until the split retainer ring is exposed and removed, the faulty nut will then pass over the male flange for removal. Replacement of a spare nut would be the reverse of this procedure.

The stub ACME thread is recommended as one way to reduce thread failures. It is almost impossible to cross thread and is easy to start even if not visible to the operator. Another consideration in fitting design is to have the seal material made of a softer material than the seal cavities to preclude damage to the welded-in flanges. Another concept would be to have a backup o-ring as a redundant seal to the metal seal. The o-ring would be made of Teflon or other compatible material.

The main reasons for the decreased usage of hoses is their low reliability. Failure histories reveal hoses are permeable (pin holes and cracks) to both fluids and gases. Flexible lines are being introduced in increasing numbers because they are not prone to leakage. Another appealing feature of flexible tubing is that it weighs less than hose. For example, titanium tubing, such as 3AL-2.5V weights approximately 70% less than Teflon hose when comparing ½ inch sizes and 80% less than Teflon hose when comparing the 1 inch sizes (Ref. 1).

b. *Burst/rupture* - Failure reports indicate that many operational problems relate to parameters which are not covered by standard test methods. Failures, as generated in burst, impulse, or flexure testing, are rarely reported in service.

The following information was extracted from a North American Rockwell Corporation report By C. W. Halsley (Reference 2).

"1) In a high-performance hydraulic system, plumbing failures are almost inevitably bending fatigue failures. Impulse fatigue failures occur occasionally and burst failures almost never.

"2) Bending fatigue failures occur as a result of two factors;

a) Low-frequency, large-amplitude flexure of the structure to which the tubing is attached,

b) High-frequency, low-amplitude vibration, which causes certain portions of the tubing system to resonate, creating high localized bonding stresses.

"3) Bending fatigue failures occur most frequently at rigid tie-off points such as bulkhead penetrations, valves, actuators, etc. They seldom occur in or near midline unions.

"4) The specific location of these bending failures is usually at (or near) the point where the tubing enters the fitting.

"5) All other things being equal, the lowest volume and lightest weight plumbing system will be that system which incorporates tube-fitting joints which have the highest possible bending endurance limit stress.

"6) ...A greater portion of the tube routings should have a complete stress analysis. Account should be taken of the differential motion of end points and support points as well as of differential thermal expansion, the sliding friction in support clamps, and pressure effects."

It can be seen from the foregoing list that a flexural endurance test is the basic test in any plumbing test program. The impulse test is a distant second. If a plumbing system is to be evaluated for minimum weight, the data must be presented in S-N curve form (stress versus cycles of stress reversals for failure).

c. Martin Data (Titan Problems and Solutions) - All lines and fittings used on the three hydraulic subsystems of Titan III are the MS type, chosen for ease of installation and component replacement and control of contamination and leakage. The lines and fittings connect the hydraulic components and carry hydraulic fluid throughout the system. All lines and fittings are designed to operate at 3000 psig, at an oil temperature range of -55°C to +135°C. The transtage has an additional safety margin because of the reduction of system pressure to 2400 psig and changing the low temperature requirement to -29°C.

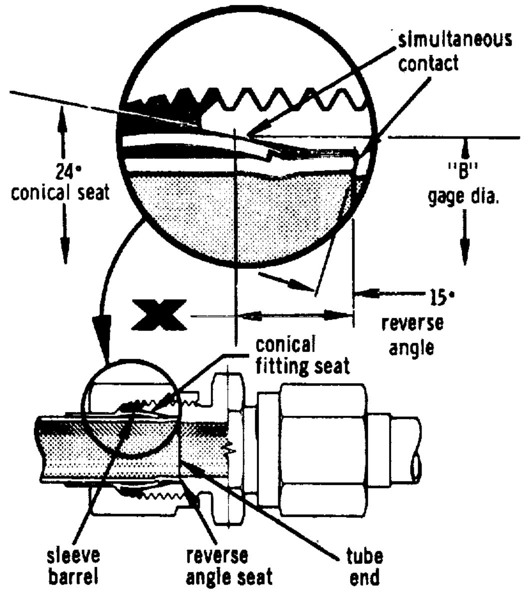
The same materials are being used on Titan III as were used on Titan II. All the hydraulic lines are stainless steel; sleeves are cadmium-plated carbon steel; and B-nuts and fittings are

aluminum. Carbon steel sleeves are used instead of stainless steel for improved presetting properties. Aluminum B-nuts and fittings are used because of the prohibitive weight of stainless steel. A polyolefin boot is shrunk over all aluminum B-nuts and fittings, for propellant compatibility. A tool has been designed for easy removal of the polyolefin boot if a component replacement is necessary.

MS fittings and lines have been used since the beginning of the Titan I program. Two problems were encountered during Titan I usage. First, B-nuts incorrectly heat treated by the vendor were found to be subject to stress-corrosion cracking in the salt spray environment at the launch site. Correctly heat-treated B-nuts are not subject to stress corrosion and, in addition, the polyolefin boot used in Titan II for propellant compatibility affords protection against this environment.

The other problem caused a flight failure of Titan I Missile J-2. Just after liftoff, a hydraulic fitting separated, resulting in loss of hydraulic pressure and destruction of the missile. The fitting was recovered and failure analysis revealed that the sleeve on the line was not preset properly, allowing the line to pull loose during a system pressure surge. As a result of this failure, an intensive study of tubing fabrication and installation was made, from both a design and a manufacturing standpoint. Reinspection of existent tubing revealed other tube assemblies out of X-control and inadequate presetting of sleeves. X-control is a method of gaging that provides a consistently accurate sealing cavity and guarantees simultaneous tube bottoming and sleeve contact with the fitting (Figure 1). With X-control, optimum spring loading of the tubing sleeve occurs when final torque is applied. In production, the use of X-control eliminates the need for variations in sleeve setting to accommodate variations in cavity depths and conical seat angles of fittings.

For better X-control, a new Weatherhead presetting machine was installed that is capable of producing uniform preset tube assemblies. New gages were designed and put into effect. In addition, a daily inspection was established for all tooling and gages. All out-of-tolerance parts are immediately scrapped and replaced. Pressure test fittings are also inspected and replaced when necessary on a similar basis. For control of presetting and assembly, manufacturing process sheets were reviewed and revised where necessary



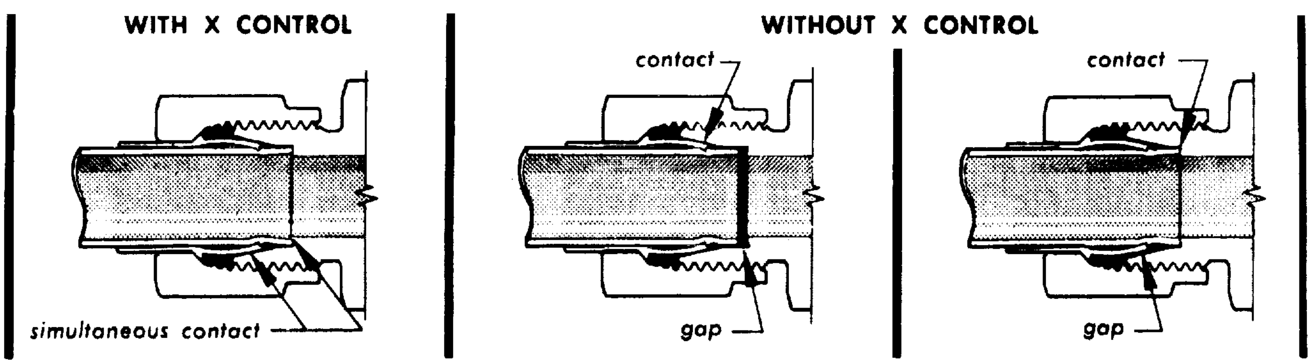


Figure 1 X-Control

to establish an exact method for presetting sleeves. An education program was given to tube fabrication, inspection, and QC personnel to show how to properly present MS sleeves and check for quality of the finished assembly. All tubing installations were reviewed for proper tiedown and clamping.

No repetition of these problems has occurred on Titan II. During Titan II use, only one tube assembly has been rejected and this was for leakage.

d. *Failure detection methods* - The methods used to detect fittings failures include: visual; go, no-go gages; dye checks; x-ray; ultrasonic; system tests prior to flight and system measurements during flight. The methods used to detect tubing and hose failures include: visual, x-ray, ground checkout tests, and system measurements during flight.

e. *Solutions* - The trend among aircraft and aerospace companies is toward eliminating reconnectable type fittings and hoses. For example, a comparison of the McDonnell Douglas DC-8 with the DC-10 reveals the following trends (Ref. 3):

1) The length of tubing has increased 53%,

2) 70% of the B-nuts have been eliminated,

3) Brazed or swaged joints are used in favor of B-nuts by a factor of 4 to 1,

4) The use of flexible tubing has increased

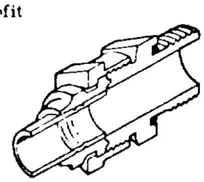
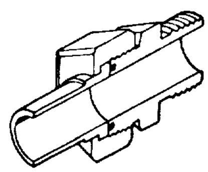
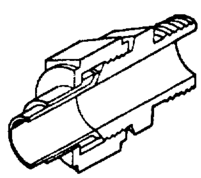
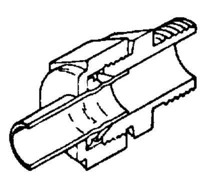
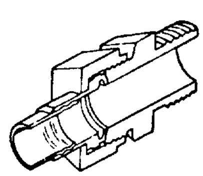
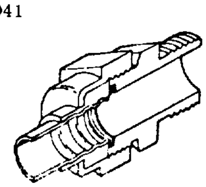
5) The use of flexible hose has decreased,

6) 97% of the "o" ring seals have been eliminated by the use of non-biting metal lip seal.

2. Design

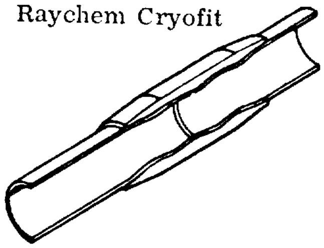
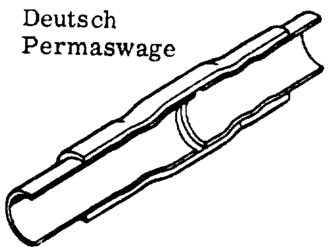
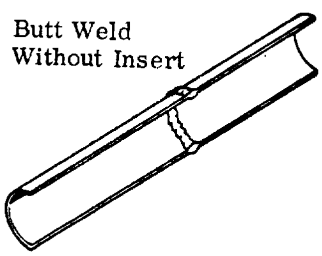
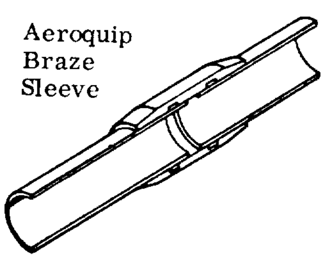
a. *Selection Criteria* - This section covers the various fitting designs and plumbing. Tables 3 and 4 compare several types of separable and permanent types of fittings. These in-depth tables cover attachment to the tubing, material, sealing methods, installation, and performance parameters. An important consideration not detailed in the table involves the time associated with fitting installation.

Table 3 Comparison of Separable Joints for 3 Al-2.5V Titanium Tube (Ref. 5)

Fitting Name	Attachment to Tube	Material	Weight of Union and Fitting	Cost	No. of Parts	Seal				Tube Prep.	Cle lin
						Type	Replacement	Protection	Leak Paths		
Raychem Cryofit 	Shrink	NiTi and 6Al-4V	Medium	Moderate	2	Integral metallic seal at tube joint	Part of fitting	Good	1	Low	Lo
CPV Min-O-Seal 	Weld	6Al-4V	Light	Moderate	3	O-Ring	New seal each reassembly	Poor	2	High	Mo rat
	Cryofit (Permanent)	6Al-4V	Medium	High	4	Same	Same	Same	2	Low	Lo
Flared 	Flaring	6Al-4V	Heavy	Moderate	2	Flared tubing against cone face of union	Part of fitting	Fair	1	Low	Low
Swagelok 	External swage	6Al-4V	Medium	Moderate	3	Curved ferrule swaged on tube interfaces with internal cone on union	Part of fitting	Poor	2	Low	Low
Resistoflex Dynatube 	Internal swage	6Al-4V	Medium	High	2	Curved washer type; lip is part of sleeve	Part of fitting	Fair	1	Low	Low
	Weld	6Al-4V	Light	High	2	Same	Same	Same	1	High	Mod rate
Harrison 400041 	Swage (could be welded)	Body 6Al-4V; seal usually gold plated stainless or inconel	Medium	Moderate	3	Removable static face seal	New seal after many reassemblies	Good	2	Low	Low

Installation					Comparable Performance					
an- ess	Equipment	Insertion Depth	Inspection Method	Cost	Fatigue Life	Collapse Characteristics	Burst Characteristics	Corrosion Properties	Field Repair
y	Tongs, Liquid N_2	Negligible	Visual	Low	Very good; tubing cracks	Good	Good	Good	Simple; need N_2 storage
le- e	Automatic tube welder and gas	Same	X-Ray	Medium	Weld fails early	Weld fails	Weld fails	Good	Poor; need weld equipment & gas
v	Tongs	Same	Visual	Low	Very good; failure in CPV	Good	Good	Good	Simple; need N_2 storage
	Flaring machine	Small amount	Measuring machine	Medium	Fair; varies considerably; base of flare cracks	Good	Good	Good	Poor; need good flaring machine
	Hand tools	Considerable	Unknown	Low	Unknown	Good	Good	Uncheckable voids	Simple; need hand tools
	Swaging tools	Negligible	Go/No-Go gage	Low	Fair; washer cracks & swaged tube fails	Unknown	Good	Uncheckable voids	Simple; need swaging tools
e-	Automatic tube welder and gas	Same	X-Ray	Medium	Weld fails	Weld fails	Weld fails	Good	Poor; need welding equipment & gas
	Swaging tools	Small amount	Unknown	Low	Fair; seal cracks	Unknown	Good	Unknown on seal material	Simple; need swaging tools

Table 4 Comparison of Permanent Joints for 3 Al-2.5V Titanium Tube (Ref. 5)

Fitting	GENERAL INFORMATION						
	Material	Profile	Cost	Wt-1/2 in. (lb)	Tube Prep.	Equipment Required	Mo
Raychem Cryofit 	NiTi	Low, smooth	Mod-erate	0.036	Low	Tongs	Sh
Deutsch Permaswage 	Commercially pure titanium AMS-4921	Low, slight ridges	Low	0.038	Low	Hydraulic swage tools	Sw
Butt Weld Without Insert 	Annealed 3Al-2.5V titanium	None	None	0	High	Automatic tube welding equipment, gas	Wel
Aeroquip Braze Sleeve 	Annealed 6Al-4V titanium plus braze alloy: 48Z-48Ti-4Be	Low, smooth	Mod-erate	0.018	High	Automatic tube welding equipment, gas	Bra

INSTALLATION						PERFORMANCE					
thod	Clean-liness	Max Tube Gap (in.)	Align-ment	Inspection	Cost	Fatigue Life	Burst	Corrosion	Est. Collapse	Est. Tensile	Field Repair
ink	Low	0.06	Not critical	Visual check of tube end	Low	Very good	Good	Good	Good	Fair	Simple; need liquid N_2 storage, hand tool
ige	Low	0.30	Not critical	Go/No-Go gage	Low	Quest-tionable	Good	Stress corrosion suscep-tible	Good	Fair	Simple; need hand/ hydraulic tools
d	Mod-erate	0	Critical	X-ray, dyecheck	Mod-erate	Poor	Poor	Good	Poor	Good	Poor; need welding equipment, gas
ze	High	0.35	Critical	X-ray, ultrasonic	High	Fair	Good	Unknown on braze alloy; good on 6-4	Good	Fair	Very poor; need cleanliness, braze equip-ment, gas

Analysis (Reference 4) of the time to assemble and inspect sundry joints were estimated to be:

Type	Time (Minutes)
Mechanical	15.15
Welded	41.72
Brazed	5.22
Swaged	2.0

The trend in fluid and gas systems is to use as many permanent joints as possible. Special manufacturing methods are provided by assembly of components and tubing into clusters on the bench prior to vehicle installation.

1) Separable fittings - In the analysis of fittings, if it is necessary to have in-flight maintenance capabilities, the system must have separable mechanical fittings. Separable fittings are required even though they have disadvantages of greater weight and have a greater potential for leakage than brazed, swaged, or welded joints. Each of these couplings consist of a threaded flange, a nut, a flange and seal as shown in Figure 2. This is a Gamah fitting. The tube is swaged to the fitting, a reusable metal seal is used and ACME threads are used for this design. The primary difference in the various separable joints is the seal configuration. Other design variations include: stub ACME thread, rather than the straight thread, and seals.

Reference 6 indicates that unskilled personnel can easily make a joint that will meet a 10^{-4} atm cc/sec leakage rate. And more emphasis should be placed on the accessibility to the fitting, the case of assembly/disassembly of the fitting and repeated high reliability with each assembly.

Table 5 presents the desirable features of separable fittings.

2) Permanent fittings - There are basically three different joining methods used for attaching fitting sleeves to tubing in affecting a permanent attachment of tubing (these methods are also used for sleeve attachment to tubing on the separable joints). The joining methods are: (1) brazing, (2) swaging, and (3) welding. These fittings are used in system locations where routine disconnections of the plumbing are not required or planned.

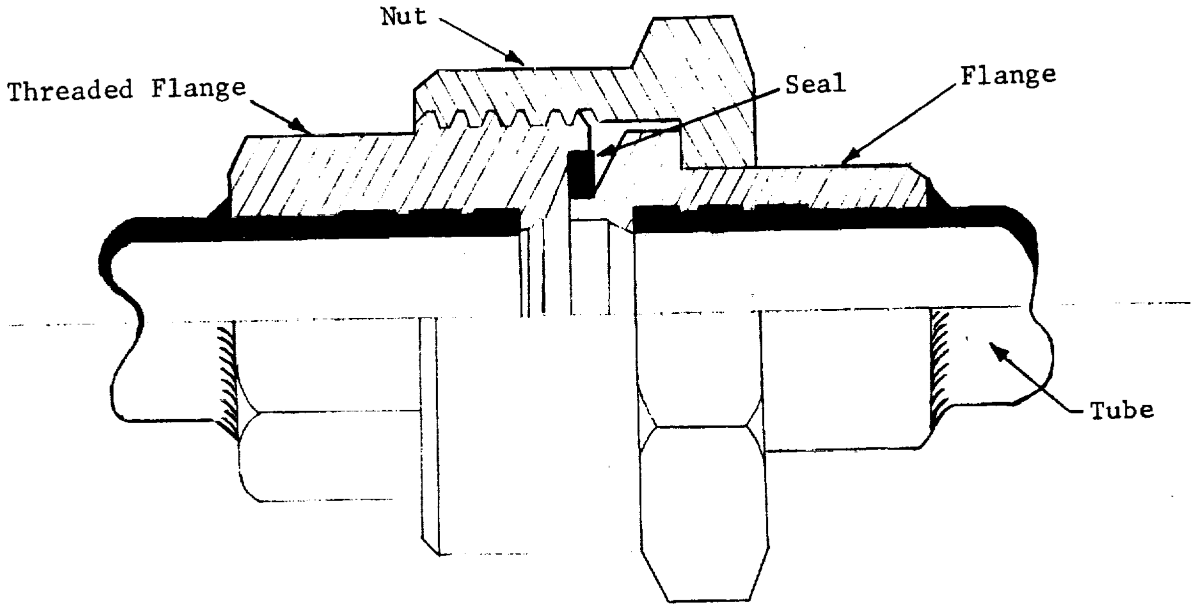


Figure 2 Gamah Fitting

Table 5 Applications and Advantages of Separable (or Reconnectable) Fittings

1. Are necessary for replacement or field repair of failed components.

2. Desirable features of current separable fitting designs are:

    1) Seals (soft or Metal) where fitting efficiency increases with higher pressures;

    2) Significantly lighter than flared, modified flared, and flareless fittings;

    3) Stronger structurally than AN and MS fittings;

    4) Requires a small installation envelope;

    5) An anti-creep capability, which maintains leak-proofness through full temperature range under all conditions of thermal cycling;

    6) A minimum tube shoulder protrusion into the connector or flat butting. This facilitates installation of plumbing, particularly short runs and straight lengths;

    7) Because the stress on sealing surfaces is controlled by the dimensions built in at the factory, the fitting designs are less torque sensitive.

    8) Those which do not require a fine machine finish on the sealing surface (up to 250 rms) have been sealed successfully;

    9) Anti-creep design which doubles as a self-locking device. This is due to high torque resistant structural design and materials;

    10) Contoured sealing surfaces that contribute to the protective characteristic of fitting design. In addition, hardened high strength steel construction affords further protection;

    11) Capable of repeated sealing integrity after numerous assembly and disassembly operations in zero gravity;

    12) Foolproof in assembly. It can be easily repaired or replaced in zero gravity;

    13) Those designs which eliminate or reduce failure modes.

Martin Marietta uses a induction brazing process which is a modified Aeroquip Corporation process with automatic temperature control (ATC). The ATC controls and records the brazing temperature and time cycle while monitoring the temperature of the joints being brazed with a photoelectric cell built into the ATC brazing tools. Brazing techniques include strict cleanliness requirements. The heat affected zone extends beyond the fitting envelope, reducing tube strength (Ref. 7). The desirable characteristics of permanent brazed fittings are presented in Table 6.

An Astro-Arc automatic welding process joings tubular members together by gas tungsten-arc (GTA) welding. This process is an inert gas shielded, metal-arc method using a non-consumable tungsten electrode. The inert gas is argon. The weld is accomplished by fusing material with a tungsten arc without adding filler wire (Ref. 8). The desirable features of permanent welded fittings are presented in Table 6.

There are two types of swaged fittings; one concept uses a fitting which is expanded at low temperatures and shrinks on the tubing, the other concept mechanically swages the fitting onto the tubing. Because of the desirable features of these fittings (presented in Table 6), an important application is the fitting and tubing commercial aircraft repair capability.

3) Plumbing and hose - The first major change in hydraulic tubing material in twenty years has taken place with the usage of Armco 21Cr-6Ni-9Mn to replace AISI 304 on the new generation commercial aircraft. The 21-6-9 tubing is preferred over AM 350 because of factory preference, shop capability, and useability with standard flareless sleeves (Reference 1). Titanium tubing, because of its high strength-to-weight ratio is desirable for aircraft and aerospace applications; but it does have a stress corrosion problem.

Because of the increasing importance of weight consideration, titanium tubing is now considered the tubing of the future. Table 2 presents the current tubing and fitting applications. Because of their poor failure history the hose usage in commercial aircraft has been reduced drastically, as noted earlier. Flexible tubing is used in place of hoses.

b. Results of survey - Table 7 presents the results of the survey of part users for plumbing components and tubing. Comments from the survey are mentioned throughout the study.

Table 6 Applications and Advantages of Permanent Fittings

A. Brazed

1) Multiple-cavity bench production tooling permits brazing of 10 joints at a time for an average of 12 seconds per braze;

2) The multiple-cavity tool and the portable-powered hand tool make use of the PRODUCTION-BRAZED fittings competitive in cost with mechanically-jointed types;

3) Visual inspection of the fillet indicates that the joint is perfect, x-raying of production joints can be reduced to sampling only;

4) Brazed fittings have reduced fitting weight, 27% to 40% on aircraft systems;

5) Brazed fittings can be assembled in place in the vehicle, with pliers type hand tools or on the bench with multiple cavity tools;

6) Meets all the applicable requirements of MIL-F-18280B;

7) Guaranteed burst strength of 16,000 psi.

B. Welded

1) Equipment is relatively inexpensive;

2) Automatic programming equipment produces consistent results;

3) Fabrication on the structure is realistic;

4) Special cleanliness requirements are not required;

5) Weld integrity can be determined by non-destructive tests (radiographic, ultrasonic);

6) Weld repair can be made in most cases by a second pass over the original weld;

7) Industry confidence in tube welding has reached maturity.

C. Swaged

1) Reduced weight and size over AN/MS fittings;

2) No special cleanliness requirements;

Table 6 (concl)

3)	No x-rays are required. The integrity of the joint is checked with a simple "Go"/"No GO" inspection gage'
4)	The fitting is mechanically swaged to the tube. No compensation in wall thickness is required from annealing of the tubing and fitting;
5)	No annealing such as in welding or brazing;
6)	Dissimilar fitting and tube materials can be joined. Stainless steel fittings on titanium tube are acceptable. Identical tooling is used and swage tooling is simple.

Table 7 Survey of Part Users

Part Type (Category)	User	Unique Test or Specification Requirements	Rationale/Justification
Fittings/ tubing	McDonnell Douglas Corporation	1. Type of joint, 2. Tube material, 3. Installation.	Brazed joints and Permaswage-type swaged joints are utilized in the DC-10 tubing system, as well as MS-flareless fittings where required for reconnectable joints. Titanium heavy wall coiled tubing is used for flexible joint system connections. Clusters of tubing runs are braze-assembled on the bench and joined together in the aircraft by Permaswage-type joints. In-situ swaging removes preloading on the system once installed, but it must be clamped so as to assure no additional stresses are imposed.

Table 7 (cont)

Part Type (Category)	User	Unique Test or Specification Requirements	Rationale/Justification
Tubing	Boeing Aircraft Company	Tubing specifications allows 5% ovality; SST tubing objective was 3% maximum.	Calculations indicate that an ovality of 5% can introduce as much as 50,000 psi stress in the tube. Tube material quality is an important factor. A passivated surface helps to reduce corrosion. Initial deliveries of tubing material for the 747 were 100% ultrasonic inspected; the confidence gained from the procedure now directs spot checking for quality requirements. Fitting and sleeve quality is important; Boeing purchases these items to their BAC specifications instead of the MS21900 series requirements.
Tubing	Resistoflex Corporation	Titanium tubing must not exceed 2% ovality.	Supported the critical effect of excessive ovality on reducing the service life of pressure system tubing. Their testing indicates that titanium tubing ovality must not exceed 2%.

Table 7 (cont)

Part Type (Category)	User	Unique Test or Specification Requirements	Rationale/Justification
Tubing/hose	Lockheed Missiles and Space Company	Minimum use of flexible hose was a criteria. Tubing alignment - critical parameter.	It was agreed that the original installation of the tubing system and the units with specific reference to routing and alignment was most important towards assuring minimum in-service problems. Clamping the tubing into place without stress is a definite requirement in the production installations.
Fitting/ tubing	Lockheed Missiles and Space Company	For L-1011 usage welded tubing butt joints were used to assemble groups of components.	Design studies associated with the SST 4000 psi system and experience with the C-5A system plumbing were major factors for the decision to use welded butt joints. Much of the system joints are welded in-situ; for this approach, welding was determined to be more reliable than brazing.

Table 7 (cont)

Part Type (Category)	User	Unique Test or Specification Requirements	Rationale/Justification
Tubing/ fittings	Martin Marietta Aerospace	Each piece of tubing is proof pressure tested. Subassemblies are pressure tested. Limited amounts of weld and braze joint rework.	Verification of tubing strength and integrity. One reweld can be attempted on a leaking weld joint (determined by proof pressure test). Three reheats can be attempted on brazed joints. Defects are determined by operating pressure tests and 100% x-ray (two planes) of each brazed joint.

Table 7 (concl)

Part Type (Category)	User	Unique Test or Specification Requirements	Rationale/Justification
Tubing/ fittings	Grumman	Tubing material requirements.	The tubing selected for the F-14 hydraulic system is;3AL - 2.5V cold worked, stress relieved, and glass bead peened titanium. A change to the annealed condition is contemplated because of difficulty in fabricating tube bends. This will result in a weight increase of about 40 lbs if cold worked titanium is used; however, if CRES steel were used the weight increase would be about 250 lbs. Gamah fittings, brazed, and welded fittings are used.
		Gamah fittings which are swaged onto tubing.	Metal seals are re-usable good leak-tight characteristics, and good seals can be made with a 32 rms finish.

c. Alternate approaches - Recommendations for development of long-life components and research into state-of-the-art advances for long-life assurance include:

1) Maximize the use of permanent joints;

2) Maximize the use of titanium tubing;

3) Develop standard separable (reconnectable) fittings to replace the current MS and AN fittings;

4) Develop tests which duplicate service reports and problems;

5) Obtain industry agreement on common test and design criteria.

d. Hardware life - Although long-life capabilities of tubing and fittings have not been demonstrated by spacecraft, these components have proven 10-year life minimums on commercial aircraft and Titan missiles.

e. Application guidelines - Although fittings are not interchangeable, different types are used in the same system.

Maintainability is accomplished by:

1) Using coupling replacements with long flanges;

2) Coupling with replaceable nuts;

3) Splices with weld-braze repair sleeve;

4) Leak repair kit.

Reference 9 describes a program which examined many different sealing techniques. The program conducted tests during the repair of leaks in tubes, fittings, and other components. The effort was directed towards a technique that was applicable for spacecraft use. Many sealing techniques were dropped from consideration because of obvious disadvantages. Tests were conducted on the remaining candidate techniques, and a comprehensive series of tests were conducted for the most promising method.

It was concluded that the simplest and most adaptable method of sealing leaks was a process using anaerobic adhesive followed by overwraps of tape using the following procedure:

1) Apply a primer to the metal surface;

2) Apply anaerobic adhesive to the leak area;

3) Wrap the area with self-vulcanizing silicone tape;

4) Overwrap with pressure sensitive teflon tape;

5) Allow the adhesive to cure prior to resuming system operation.

The above method is simple, can be accomplished by an inexperienced person, requires no special tools, and can be performed in less than five minutes. The only constraint is that the system must be shut down and depressurized prior to sealing the leak. For most systems, this is normally considered to be a standard operating procedure anyway.

It is recommended that further development be performed on leak repair technology capabilities in space.

D. TEST METHODOLOGY AND REQUIREMENTS

Qualification testing of plumbing components and tubing include: Vibration, pressure, tensile, collapse, flexure, and corrosion testing. Accelerated and life test data are determined by S/N fatigue life characteristics as described below.

It is recommended that testing techniques be developed which closely simulate actual usage conditions, such as those listed below. SAE committees have closely monitored advanced testing for the last few years.

Service reports indicate that many failures relate to areas which are not covered by standard testing. For example, failure modes in burst, impulse, or flexure modes are very rare in service; instead, the major portion of reported incidents are related directly or indirectly to tubing installation, tubing fabrication, and line supports.

Reports and observations of this nature led to a new approach of combined environmental evaluation testing on complete systems or subsystems. This approach emphasizes the complete systems exposure to operational environments. It was developed to assess the validity and justification of presently used qualification performance, and environmental tests. Another reason was to develop basic criteria and advanced test techniques because of the introduction of new materials, components and advanced installation technology.

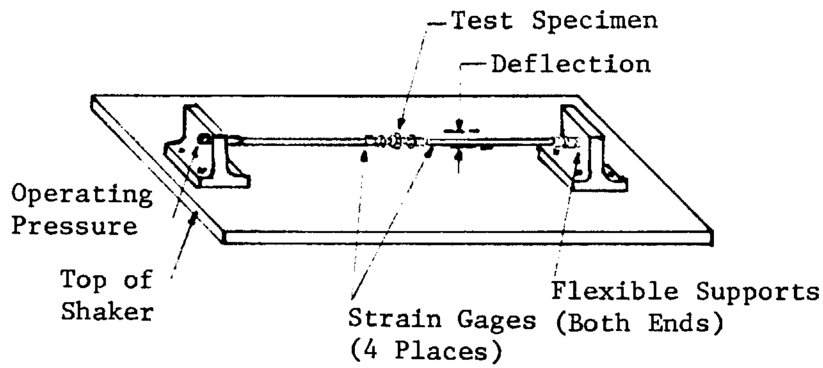
The need for clarification of flexure test requirements in MIL-F-18280 has led to the development of the proposed new recommended Practice ARP 1185 (Reference 10), Flexure Testing of Hydraulic Tubing, Joints and Fittings. Advanced flexure testing methods described in ARP 1185 (planar or rotary cantilever bending, or planar free-free beam method) are summarized below:

1) Rotary or Planar Cantilever Flexure:

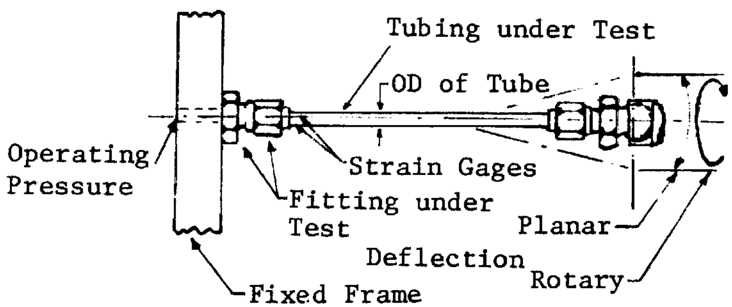
   This test is intended for conducting flexure tests of new and/or existing tubing materials, fittings and joints. A constant mean stress is applied by holding system pressure in the test specimens. Typical Tube-Joint Test Methods are shown in Figure 3.

2) Planar Vibration Free-Free Beam:

   This test is used to establish the relative flexure fatigue strength of new tubing materials and fitting joints.



Flexure Test - Simple Beam Method



Flexure Test - Cantilever Beam Method
per MIL-F-18280 and ARP 1185

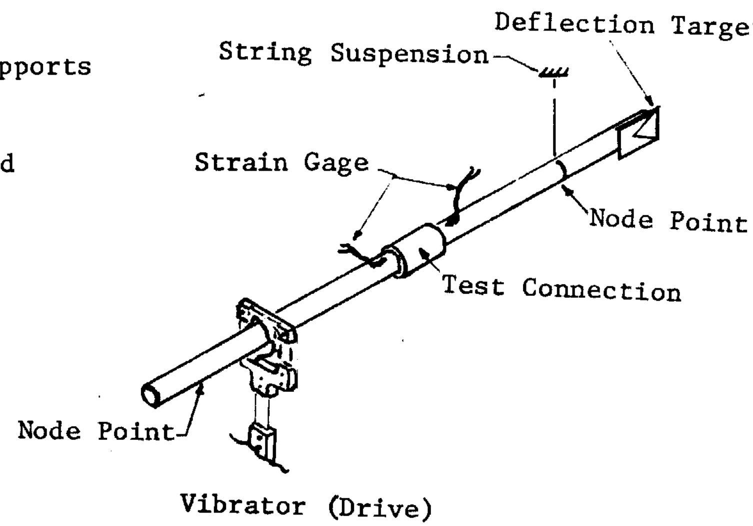


Figure 3 Typical Tube-Joint Test Methods

Since these components do not have moving parts, burn-in or wear-in tests are not performed. However, pressure tests are used to verify integrity of tubing and fittings in the system.

E. PROCESS CONTROL REQUIREMENTS

Preparation for brazing and welding operations include removal of burrs and sharp edges (internal and external .005" max). The wall thickness tolerance shall be ±10%. Accomplish brazing in a Code 3 clean room.

In case of an unacceptable brazed joint, the number of joint reheat cycles shall be limited to 3 times to affect a good joint. In case of an unacceptable weld joint, the repair procedure shall include:

1) Only one re-weld attempt;

2) Cut out the weld bead, insert a repair sleeve and re-weld in two places;

3) Use of longer weld inserts or brazing is permitted.

Torque schedules are necessary for separable fittings. Require Teflon seal changes after approximately 50 reassembly times. Metal seals require a change after approximately 3 to 12 reassembly cycles.

Excessive tube ovality reduces the service life of system tubing. It is recommended that the tubing specification limit ovality to 3% maximum.

Stains and pits have been found on the internal surfaces of stainless steel tubing on the Skylab program (Alert Number MSFC-71-20). The corrosion is generally located in the center of the tube making visual inspection difficult. Pits up to .007 inches deep have been reported. It was determined that this condition was caused by inadequate control of pickling, rinsing and/or drying procedures at the tube manufacturers.

Solvent vapor degreasing does not appear to adequately clean the internal surfaces of small diameter tubing since heat transfer to the external surfaces allows no condensation of the solvent internally. The following procedures which are recommended for future programs have been instituted to detect and solve the problem on Skylab:

1) One hundred percent borescope visual inspection has been set up for receiving inspection;

2) Future orders for tubing will require that tubes be rinsed with demineralized water and forced dried immediately after rinsing.

Design Constraints

The design strengths of separable and permanent joints were presented in Tables 5 and 6. System leakage can be minimized by the use of fewer separable joints. Welded joints have represented a mature design. A brazing drawback is the cleanliness required in affecting a good joint. Swaged joints and titanium tubing will have increased usage.

F. REFERENCES AND BIBLIOGRAPHY

References

1. Mutzke, R. W.; Nelson, W. G; van der Velden, J. H.: *Future Airplane Hydraulic Distribution System Design Developments and Testing Criteria.* SAE National Aeronautic and Space Engineering Meeting, Los Angeles, California, October 1970.

2. Helsley, C. W. Jr.: *Titanium, The Material for Advanced Aerospace Vehicle Plumbing Systems.* Report No. NA-69-344, North American Rockwell Corporation, Los Angeles, California, 1969.

3. *DC-10 Hydraulics.* Report DAC-68379B, Douglas Aircraft Co., Long Beach, California, December 1969.

4. Smith, R. A.: *Investigations of High Pressure Tube and Fittings Sealing Techniques.* Report No. T-71-48844-006, Martin Marietta Corporation, Denver, Colorado, December 1971.

5. *DSSV Engineering Report, Titanium Tubing.* Lockheed Missiles and Space Company, LM SC-D177821-6, Contract N00024-70-C-0254, June 1971.

6. V. A. DesCamp, *et al: Final Report, Study of Space Station Propulsion System Resupply and Repair.* NASA Contract NAS8-25067 , MCR-70-150, Martin Marietta Corporation, Denver, Colorado, June 1970.

7. Clover, F. R. and Hurley, W. J.: *Automatic Tube Welding and Brazing.* Report No. T-71-48983-011, Martin Marietta Corporation, Denver, Colorado, December 1971.

8. Aldridge, L. L.: *Viking Propulsion Subsystems Brazing and Welding Criteria.* VER-149A, Martin Marietta Corporation, Denver, Colorado, October 1971.

9. V. A. DesCamp, *et al.: Final Report, Modification of an Astronaut's Mockup Tool Kit.* NASA Contract NAS8-26448, MCR-71-59, Martin Marietta Corporation, Denver, Colorado, April 1971.

10. SAE Committee A-6 Proposed ARP 1185, *Flexure Testing of Hydraulic Tubing, Joints, and Fittings.* May 1971.

Bibliography

1. Charhut, D. E., Byke, R. M., and McCelland, C. M.: *Design of Space Stations for Low Atmospheric Leakage.* Paper No. 71-337, AIAA/ASME 12th Structures, Structural Dynamics and Materials Conference, April 19-21, 1971.

2. Stanley, J. K.: *The Current Situation on the Stress Corrosion Cracking and Hydrogen Embrittlement of High Strength Fasteners.* Paper No. 72-385, AIAA/ASME/SAE 13th Structures, Structural Dynamics and Materials Conference, April 10-12, 1972.

3. *Study of Advanced Techniques for Determining The Long Term Performance of Components.* NAS7-782, Report No. 16568-6014-Roo, TRW Systems Group, March 1972.

4. DuBuisson, J. C., *et al.: Handbook of Long-Life Space Vehicle Investigations.* Report No. T-70-48891-012, M-70-16, Martin Marietta Corporation, Denver, Colorado, December 1970.

5. Nicol, J.: *In-Place Welding Process and Weldable Tube Fittings for Hydraulic Systems.* Paper presented at SAE Aerospace System Conference, Los Angeles, California, June 1967.

6. Robinson, N. F.: *Third Generation Brazed Plumbing.* Presented to SAE G-3 Committee, Monterey, California, July 1970.

7. SAE Committee A-6, Proposed ARP 994, *Recommended Practice for the Design of Tubing Installation for Aerospace Fluid Power Systems,* August 1971.

8. SAE Committee A-6, *Aerospace Fluid Power and Control Technologies Minutes,* April 1972.

VIII. CHECK VALVES

by P. J. Powell

VIII. CHECK VALVES

A. INTRODUCTION

Check valves allow flow in one direction and, if the system pressure reverses, close quickly to stop flow in the opposite direction. Check valves are self-contained devices, requiring no external actuation signals or sources of power. The valving elements are activated by the pressure forces of the flow media. Check valves are spring loaded and are the simplest of valve types. The types of check valves include the ball, cone, poppet, swing, and flapper. Since check valves have no external actuators, they often lack sufficient seating force to accomplish a good seal. Check valve designs are a compromise between good sealing and low pressure drops, and minimum cracking pressure.

Figure 1 shows a typical poppet check valve. The O-ring in this valve cushions the closure of the metal seats. The industry survey revealed that these valves are used in commercial cleaned systems without problems, as the poppet design is tapered to make it self-cleaning. Martin Marietta Aerospace has used this type of check valve in Titan systems without any known contamination problems. During contamination testing at Martin, the O-ring was removed from a check valve to evaluate contamination sensitivity. The valve operated satisfactorily in spite of the missing O-ring.

Another type of check valve is one in which the valve closure elements and sealing surfaces are made of an integral piece of rubber. Figure 2 shows a rubber type check valve. This configuration is designed to take advantage of the elastic properties of rubber. The seat has a series of holes, arranged in a circle, completely covered by the rubber valve. As pressure is applied in the flow direction the rubber valve element is forced away from the seat allowing the medium to pass through the holes in the valve seat. Pressure applied in the reverse direction forces the rubber element against the valve seat and checks the flow by covering the holes. A complete rubber valving element is used for low pressures. A metal back-up plate is used for high pressure applications to prevent the rubber valving element from extruding or tearing away if a high pressure transient occurs in the flow direction. This check valve is used in both a 1000 psig oxygen gas system and fluid systems on the Apollo program.

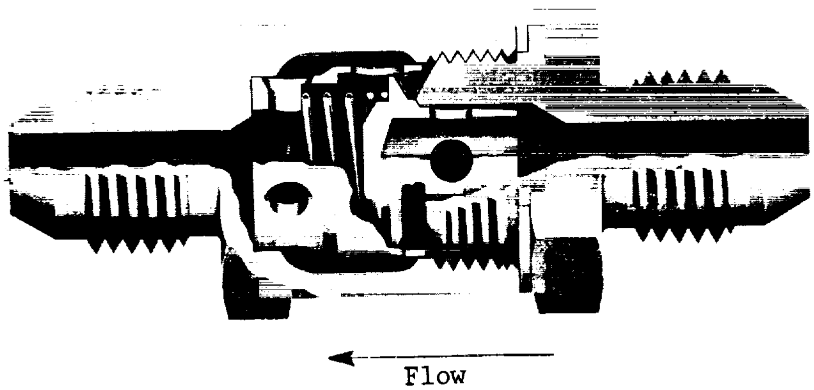
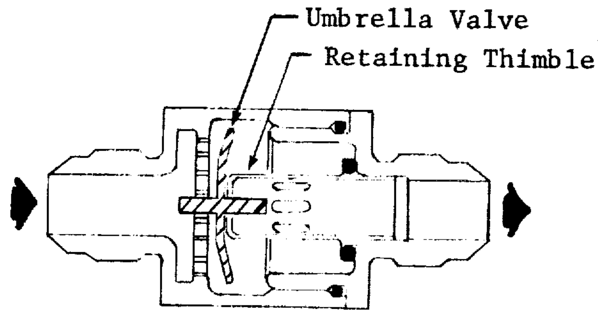


Figure 1 Check Valve



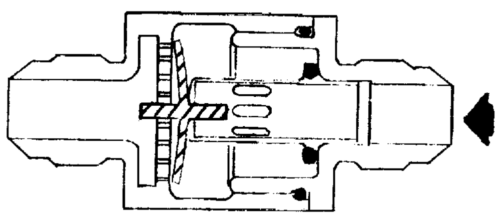


Figure 2 Check Valve

The valving element for this check valve is EPR rubber. The minimum conservative age life as shown on the Apollo program (reference 1) is ten years. The maximum estimated life under average exposure conditions is twenty years.

Two adverse features of this valve configuration have been identified: (1) The design is susceptible to contamination. However, in all the applications to date, a system filter has been located upstream of the check valve and has adequately protected the valve from contaminants; (2) The rubber element tends to flutter (chatter) at extremely low flow rates. In all cases, however, the flow rates were well below the system performance limits.

Swing and flapper check valve configurations are very similar. The swing valve uses a single circular valving mechanism in contrast to the two semicircular discs or flappers utilized in the flapper valve (Figure 3). The flapper valve is used primarily in pneumatic systems because it permits a minimum pressure drop within a given operation envelope. The flapper configuration is more desirable than the swing for long-life application because the lower travel of the flapper from the open to closed position produces less water hammer.

B. GUIDELINES FOR LONG-LIFE ASSURANCE

Check valve service lives of 10 years without maintenance are achievable. The check valve cycle life ranges from 250,000 cycles for use in high pressure systems to 500,000 cycles for use in low pressure systems. The current state-of-the-art check valves are satisfactory for programs requiring long service life, such as the Shuttle program. However, cycle life may be a problem for the 500 mission goal of the Shuttle. Preliminary analysis shows qualification life requirements of order $.5 - 1.0 \times 10^6$ cycles.

Internal check valve leakage due to contamination was the main life-limiting failure mode identified from the survey of manufacturers and users. Listed in the order of most probable occurrence, the failure modes are:

1) Internal leakage, caused by valve failing to close or stuck open, seat leakage or chattering of the poppet;

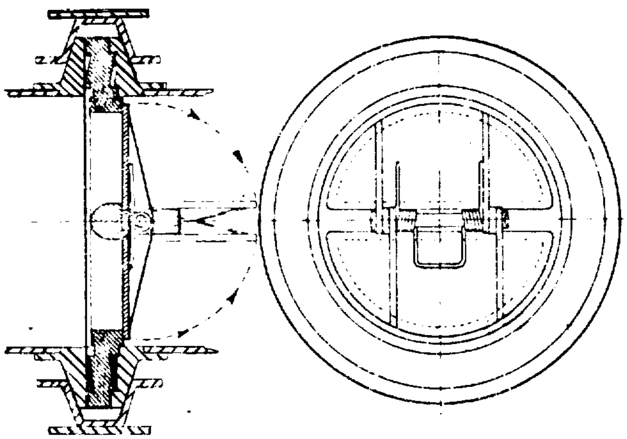


Figure 3 Split Flapper Check Valve

2) Poppet fails to open;

3) External Leakage around the static seals and through the valve body.

1. <u>Design Guidelines</u>:

1) Design large flow paths (areas) through poppet check valves and main sealing surfaces to reduce flow velocities and erosion of seats.

2) Configure valve housings (as far as possible) to eliminate areas that can entrap contaminants. Use a "swept by" design where flow will tend to pass contaminants through the valve.

3) Insure that materials are compatible with the working fluid including maximum expected fluid impurities. The evaluation must consider dynamic as well as static applications and must consider temperature, pressure, and phase variations of the fluid.

4) Minimize valve induced contamination:

   a) Material selection should consider the effect of wear particle size on useful life. Particle sizes vary as the Young's Modulus divided by the square of the compressive yield stress.

   b) Use rolled threads in preference to machined threads to minimize burrs and achieve higher strength.

   c) Minimize dead-end passages and capillary size passages.

5) Leaks due to contamination are minimized if the seat load is sufficient to plastically yield a trapped contaminant particle on the sealing surface. System contaminants must be identified to determine particle size and material properties.

6) Avoid sliding parts; they not only induce contamination, but entrap contamination.

7) Develop valving elements with quick opening areas to preclude chattering, instability and high seat velocities.

8) Do not use ball check valves as they tend to chatter, particularly on rapid closure of the valve element.

9) Spherical or tapered poppets are recommended. These poppets are self-aligning, but require precise alignment of the valve seat and the poppet assembly.

10) Insure guiding of poppet onto valve seat to allow for maximum alignment and eccentricity tolerance (i.e., use a large length/diameter bearing surface to guide ratio - minimum 2:1).

11) Justify not employing elastomer seats. Low leakage rate requirements (less than 5 scc/hr) can be most easily met by elastomer seats. Since low seat stresses are desirable for long life, high safety factors should be used in conjunction with high seat loads. Control the undesirable creep or cold flow characteristics of Teflon seats by:

    a) Containment of Teflon seats on three sides;

    b) Heat stabilization techniques;

    c) Low stress levels.

12) Avoid Teflon seals for high pressure oxygen system applications to preclude flammability hazards.

13) Seat stress of plastic seats must be held well below material yield stress for long-life storage to prevent excessive deformation. Stress in elastomeric seals are generally not critical.

14) Wear is minimized by making the hard seat in the housing wider than the plastic seal of the poppet or conversely by making the metal sealing surface of the poppet wider than the plastic seat.

15) Smooth surface finishes are superior for long life; therefore, it is recommended a finish of 16 - 32 rms be used.

16) Lipseals supported with a back-up ring to resist undesirable influences of pressure loading are superior to designs without this provision. It is recommended that lipseals guiding piston rods or poppets use backup sealing rings in addition to the main lipseal.

17) Avoid plastically deformed soft metals. This configuration has excellent sealing characteristics, but exhibits low life.

18) Do not use large flat plastic poppet sealing interfaces. They are not compatible with long-life due to high impact wear characteristics and misalignment.

19) Provide positive stops at the end of travel to minimize transient stresses due to poppet travel.

20) Seal retention methods shall prevent seal distortion, creep or dislodgement.

21) The inherently larger leakage rates of hard on hard seat configurations can be minimized by lapping poppet and seat to obtain better seat finishes.

22) Reduce life due to vibration sensitivity is minimized by decreasing available clearances in bearings and guides, avoiding large overhung moments, and restraining lateral motion of poppets.

23) Provide external fittings to permit drain, flush and purge after exposure to propellants.

24) Control stress corrosion by avoiding stress corrosion susceptible materials and/or design parts to operate at low stress levels.

25) Attempt to eliminate dynamic seals, They are subject to wear and cause unpredictable drag.

26) Control external check valve leakage by:

    a) Requiring welded valve body construction;

    b) Require vacuum melt bar stock;

    c) Impregnate valve casting with sealant.

2. Process and Control Guidelines

1) Ultrasonically clean valve parts; assemble in specified level-clean areas; and govern by a single contamination control specification during test. This same control specification should also govern the test fluid media used.

2) Use a fabrication barrier (bag) to protect clean parts. Consider using nylon or polyethylene to prevent creation of contamination due to chaffing of the barrier by the parts.

3) Vendor controls shall guarantee that the valve contamination particle size and count will not exceed specified limits. Documentation is required.

3. Test Guidelines

1) Conduct contamination susceptibility tests during development to determine the level of contaminant that the valve can tolerate.

2) To verify valve operation conduct 50 to 100 run-in cycles.

3) Do not rapid cycle valves designed for fluid applications for functional verification in a dry condition because the lack of fluid damping can increase seat stress and reduce life.

4) Consider techniques to stabilize cold flow. Initial analysis of an accelerated test for Teflon valve seat poppet assemblies indicate that if the assembly is subjected to normal design loads and 150°C temperature for 14 days, then the assembly would not experience any more appreciable cold flow in a ten year period.

5) Conduct life cycle endurance tests under operational conditions. For long-life applications, the valve cycle life parameter must be known.

4. Applications Guidelines

1) A solution to long-life and reliability problems is the application of check valves into redundant configurations. Examples: Several companies have developed either a series parallel check valve (quad valve) with integral filters or a series type check valve with double seats.

2) Install valves into functional groups within systems using permanent connections, such as welded or brazed connections to avoid contamination and to prevent leakage. This type of installation would allow a group of valves to be replaced in event of a malfunction.

3) Check valves cannot normally be repaired by inflight maintenance procedures.

C. LIFE LIMITING PROBLEMS AND SOLUTIONS

1. Failure Mechanism Analysis (FMA)

This analysis is summarized in Table 1.

a. Failure Modes - The check valve component failure modes are discussed below in the most probable order of frequency.

1) *Internal Leakage* - The failure mechanisms contributing to internal leakage are poppet failing to close, seat damage, and instability or chattering of the valve poppet. The most prevalent failure mechanism noted in the survey of manufacturers and users was the valve failing open due to contamination between the poppet stem and the valve body. The effect of this failure is internal leakage past the valve seat. Contamination may be contained in the system medium, may be generated by other components in the system, or may be generated by the component. In addition, contamination may be induced into the valve by the manufacturing processes. Chattering, or uncontrolled cyclic action of the valving element, can be described as the hammering of the poppet against the seat at high frequency. This phenomenon causes pitting, wear and abrasion.

2) *Poppet Fails to Open* - Contamination is the most prevalent cause of this failure mode.

3) *External Leakage* - External leakage can occur due to leak paths through static seals or valve body porosity.

b. Failure Mode Detection - Check valve malfunctions are inherentry difficult to detect since the valves are completely encased and are operated by internal system demand rather than external stimulus. Malfunction detection is generally dependent upon subsystem performance evaluation and fault isolation. Built in Test Equipment (BITE) can be incorporated in the valve for component functional verification tests prior to flight. A method to check out redundant check valves, by the use of test ports is shown in Figure 4. This is a time consuming task and is limited to ground operations.

Table 1 Failure Mechanism Analysis Check Valves

PART AND FUNCTION	FAILURE MODE	REL. RANK	EFFECT ON VALVE	FAILURE MECH.	DETECTION METHOD	HOW TO ELIMINATE/ MINIMIZE FAILURE MODES
Spring loaded poppet - (control flow)	Poppet fails to open due to sticking and binding.	2	No flow through the valve - failure to open.	Failures are attributable to contamination.	<u>Pre-installation</u> 1) Ultrasonic cleaning and particle count. 2) Test and particle count. <u>Post-Installation</u> 1) Indirectly, by system pressure indications.	Insure guiding of poppet onto valve seat to allow for maximum alignment and eccentric tolerances by using a large length/diameter bearing surface to guide ratio-minimum of 2:1.
	Poppet fails to close, poppet chatter and seat damage.	1	Internal leakage	Failures are attributable to contamination generated by: 1) Self-induced. 2) Other components. 3) System media.	<u>Pre-installation</u> 1) Ultrasonic cleaning and particle count. 2) Test and particle count. <u>Post-installation</u> 1) Indirectly, by system pressure indications.	Filters can be placed upstream of critical components. Chattering can be avoided by proper design.
Valve body - (support valving elements and contain flow media)	Porosity of valve body and static seal leakage.	3	External leakage	Material stringers or inclusions, leakage through or around static seals.	Pressure tests prior to valve installation and indirectly by system pressure indications or visual leakage after installation.	Vacuum melt bar stock to eliminate stringers and inclusions. Castings can be impregnated with a sealant substance.

c. Solutions - Increased reliability can be accomplished by the application of check valves into redundant configurations. APCO and Parker have developed a series parallel check valve for the Apollo program. The Parker design (Figure 4) is used in the main pressurization lines for both the N_2O_4 and UDMH-Hydrazine tanks in the ascent and descent stages of the LM. The quad check valve is used to prevent back flow of helium and/or propellant in the helium pressurization system. Long-life features of the valve unit are:

1) All external features are welded.

2) No metal-to-metal moving parts in the unit; the poppet is guided by a Teflon bearing on stainless steel.

3) Teflon and Corrosion Resistant Steel (CRES) to eliminate compatibility problems.

Another Parker check valve design is used in the Apollo supercritical hydrogen and oxygen gas storage systems. This check valve (Figure 5) has double seats. At low flows the auxiliary seat is barely open and can catch contamination (if present); the main seat is wide open and protected from contaminants.

The Skylab environmental control system uses two check valves in series in the O_2 service to prevent mixture with N_2 gases.

2. Design

a. Selection Criteria - Functional elements in check valves include moving parts, such as springs, seat, and poppet. These moving parts are subject to wear after repetitive open and close actions, thus limiting the operational life of the part. Design considerations for long life assurance are identified in Table 2.

The number of cycles for a given application can be reduced by increasing the valving element spring force to dampen the oscillations. However, the upper value of the spring force is limited by specification of maximum cracking pressure.

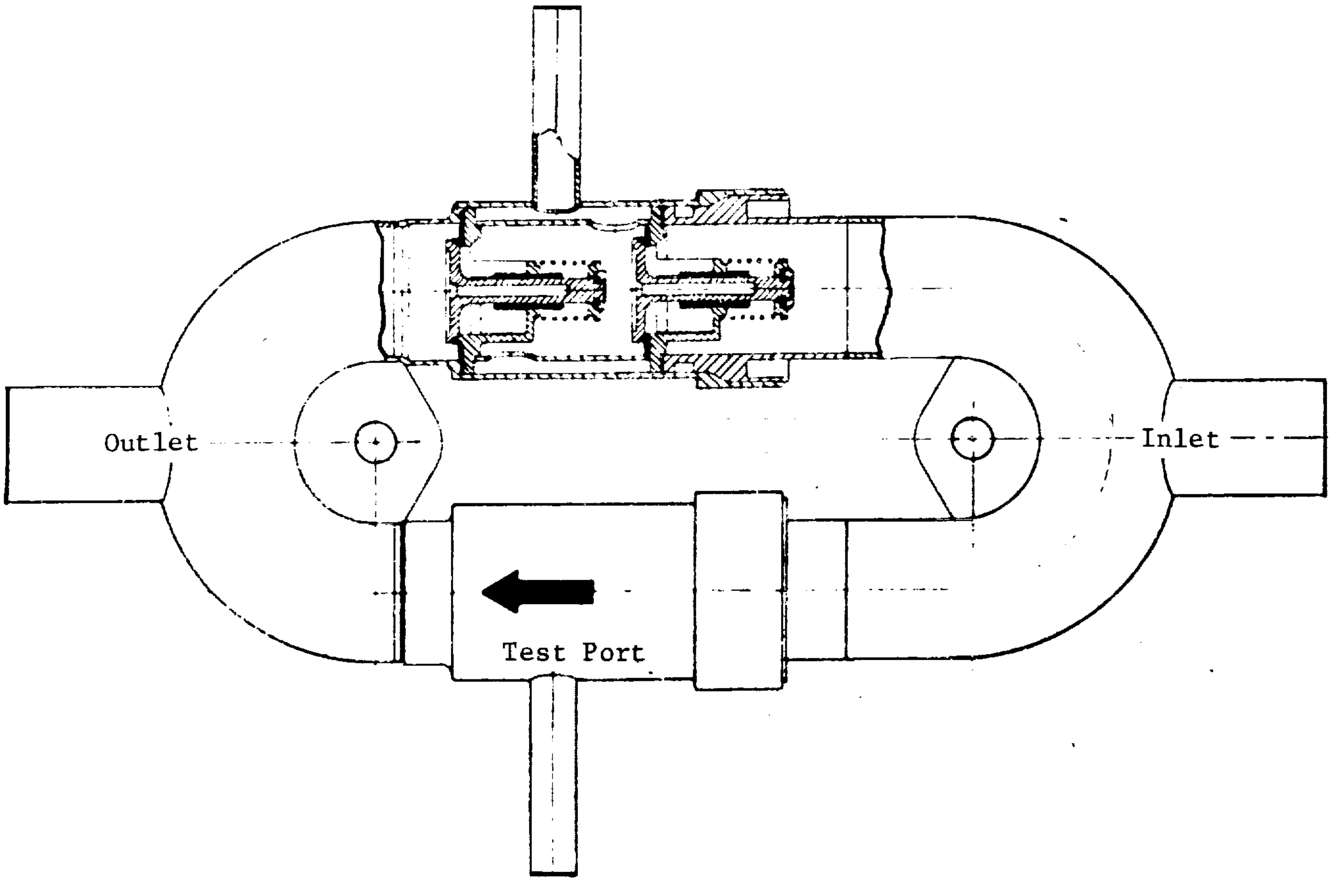


Figure 4 Series Parallel Check Valve

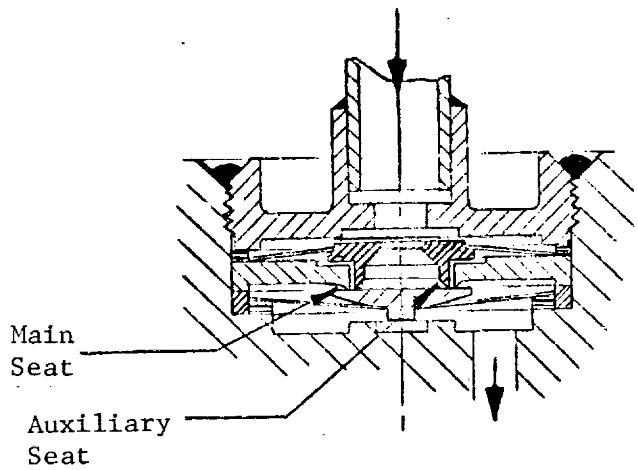


Figure 5 Double Seated Check Valve

Table 2 Design Considerations for Long-Life Assurance
Part/Component: Check Valves

Design Considerations	Discussion
Clearance Between Sliding Parts	Contamination lodged between close fitting parts in the valving mechanism can cause sticking or cocking. Eliminate areas that can entrap contamination from the valve design. Provide a length/diameter guide ratio greater than 2:1 for guiding the poppet on the valve seat. Select proper lubricant and materials for sliding surfaces.
Poppet Force Margins (open and closed)	Conservative force margins are necessary to provide sufficient seating force to accomplish a good seal and low pressure drops.
Seat Configuration	Flow around the valve seat may cause turbulence and high pressure drop. Avoid the use of ball check valve due to their tendency to chatter. Resilient seat materials are more tolerant of contamination than metal seals. Seal designs, which contain the seal on three sides to prevent cold flow of plastic seats, are recommended. Check valve differential forces must be considered in the design of valve closure and seals to prevent fracture of closure and extrusion of soft seals.
Dynamic Seals	Dynamic seals are subject to wear, unpredictable drag, and should be avoided.
Static Seals	Brazed or welded joints are preferred to captive types of seals for valve construction. Also, permanent mechanical joints, such as brazed, welded or swaged, are preferred to separable mechanical joints.
Materials Compatibility	Avoid plating of internal valve parts subject to stresses which may result in flaking, or which may scrape easily. Avoid the use of Teflon in high pressure oxygen systems or in applications where two to five percent creep, or swell, cannot be tolerated.

The industrial survey consensus revealed that better contamination control, containment of seals, and the careful guiding of the poppet on the valve seal would enhance valve life. The manufacturers listed contamination of flow media as the main cause for valve failure. Therefore, it is desirable to incorporate design features which minimize susceptiblity of the valve operation to potential contaminants and to minimize sources of contamination in the valve and the working subsystem. To minimize the effects of contamination, any area in the valve which could trap particles should be eliminated. Use of a "swept by" design allows contamination to pass through valve. Sufficient clearance should be provided between sliding surfaces such at the anticipated maximum dimension particles will pass through and not cause sticking or binding.

Threaded line connections are a source of contamination. Permanent connections are preferred to mechanical connections to eliminate this contamination source. Where threaded line connections are required, the use of rolled threads is preferred to machined threads to minimize burrs.

Resilient seal materials should be evaluated for long life application because they are more tolerant to contamination than metal seats. The advantages and disadvantages of soft seats are presented in Tables 3 and 4 respectively. Polymer selection must consider the effects of propellants, loading, temperature, and vacuum conditions. Compatibility selection details are presented in the design section of the solenoid valve report, Chapter IX.

Among the materials used for soft seats, or poppets, are plastics such as Teflon, Kel-F, nylon and mylar; and elastomers such as buna-N, silicone, and Viton.

Use of Teflon valve seats has been discouraged because of the problem of cold flow. However, the ability of Teflon to seal in an environment of particulate contamination is an indispensable advantage for long life application.

According to the survey of manufacturers and users, the control of Teflon cold flow for valve seats is accomplished by the following procedure: the valve is subjected to its normal operation pressure while maintaining a temperature of 82°C for 24 hours to accelerate the cold flow of the seat material. The data presented in the solenoid valve design section indicates

Table 3 Advantages of Soft Seats

Parameter	Advantage
Specified Leak Rate Requirements	Very low leakage is more readily achievable with plastic or elastomeric seats.
Contamination	Plastic and elastomeric seat have a greater ability to envelope particulate contamination.
Misalignment of Seat and Poppet	In larger valves, where control of misalignment becomes difficult and costly, soft sealing surfaces compensate for misalignment of seat and poppet.
Valve Size	Soft sealing valves are generally smaller, because a good seal can be achieved with less load.
Cold Welding	Where the downstream side of the valve is exposed to a vacuum, there is no concern regarding cold welding.

Table 4 Disadvantages of Soft Seats

Parameter	Disadvantage
Creep (Cold Flow)	Improperly designed Teflon or Kel-F seats may creep during long duration storage periods with the seat loaded, causing subsequent leakage.
Erosion	In high pressure applications, such as 3000 psi, seat erosion due to high velocity contamination passing a small seat area may cause leakage or gross seat failure.
Temperature	Plastic and elastomeric seats have a more limited useful temperature range (not exceeding 150°C).
Propellant Compatibility	Plastic and elastomeric seats have a more limited range of usage.
Seat Retention	Soft seats introduce the problems of adequate seal installation and retention. Fluid seats are usually integral with the valve housing.
Fabrication	Plastic seats are more difficult to machine and inspect. Manufacturing techniques must prevent welding heat input from damaging the seat.

that most of the Teflon cold flow occurs during the first 24 hours. An accelerated test should be developed for programs which require long life. Initial analyses indicate that if the valve seat is subjected to normal design load and 150°C temperature for 14 days, then the seat would not experience cold flow in a ten year application. However, metal-to-metal seats are satisfactory if the system leakage requirements can be relaxed or if rocket engine requirements include a highly reactive propellant. Where metal seats are used, leaks due to contamination are minimized if the seat load is sufficient to plastically yield a trapped contaminant particle on the sealing surface. However, for metal seat applications, high safety factors should be used to assure low seat stresses. If it is deemed absolutely necessary, filters can be placed upstream of critical components. One company has installed integral filters in their check valves, to minimize contamination effects.

The possibility of valve mechanism binding must be minimized to prevent excessive wear. To achieve this goal, clearances in valving mechanisms should provide for thermal cycle expansions and/or exposure of non-metals to fluid. A length/diameter bearing surfaces guide ratio of greater than 2:1 is required to guide the poppet on the valve seat.

Conservative force margins are necessary when developing cracking and reseating pressure characteristics to provide sufficient seating force to accomplish a good seal and low pressure drops. The maximum check valve differential pressure in the reverse direction must be considered in the design of the valve closure and seals to prevent fracture of closure and extrusion of soft seals.

Chatter can be minimized by utilizing valving elements with quick opening areas. The following example illustrates how a failure mode attributed to poppet chatter was eliminated.

Mr. Fred Wright, Martin Marietta Aerospace, noted that a check valve used in gaseous nitrogen media, developed for the Mariner '71 propellant subsystem, experienced poppet chatter. Increasing the friction of the valve poppet stem was tried to dampen the chatter, but the cracking pressure was not repeatable. The poppet chatter was eliminated by opening the clearances on the flow passages. This caused the poppet to lift and resulted in a longer poppet stroke when the valving element was opened.

The TRW handbook (Reference 2) indicated that valve chattering may be reduced by:

1) Diminishing the difference between poppet projected area and the orifice area.

2) Adding a dashpot effect in the spring cavity.

3) Using loading springs with negative spring rates to get snap action opening (belleville springs).

Ball check valve designs should be avoided because they tend to chatter upon rapid closure.

Dynamic seals are subject to wear and should be avoided for long-life applications. The Martin Marietta Aerospace has attempted to avoid sliding surfaces in developing valve designs because of wear and contamination problems.

An important performance characteristic is the valve seat configuration. Flow around the valve seat may cause turbulence and seal erosion due to contamination passing at a high velocity. Spherical and conical valve poppets are recommended. More information is presented in the solenoid valve chapter.

The methods suggested to eliminate external leakage are to use welded valve body construction and permanent connections such as welded, brazed, or swaged rather than reconnectable mechanical joints when installing valves into the system. The valve bodies can be machined from bar stock, cast, or forged. Martin Marietta has had material problems with check valve body porosity. The valve bodies were machined from 347 stainless steel bar stock and the material had stringers or inclusions. This problem can be solved by selecting a consumable-electrode-melted bar stock (vacuum melt) to eliminate inclusions.

Complex valve body design can be sand cast; however, they are relatively weak and may be porous. The porosity can be controlled by impregnation of the casting wall with a sealant substance such as sodium silicate.

Forgings generally weigh more and require more final machining than castings. However, forgings are structurally stronger than castings and able to withstand higher internal/external loading.

If leakage is a stringent requirement, then brazed or welded valve body joints are preferred to captive types of seals for valve construction.

Aluminum alloy valve bodies can be conditioned for corrosion resistance by providing a chromic or sulphuric acid anodized coating. CRES alloys can be used to fabricate steel alloy valve bodies. Stress corrosion cracking is caused by material under a tensile stress while on a corrosive atmosphere. It is recommended that protective coatings be used on materials that are subject to stress corrosion, or that are exposed to corrosive environments.

b. Results of Industry Survey - A survey of check valve manufacturers and users was conducted to ascertain the expected life of existing hardware and to identify features which would enhance valve life in new designs. The consensus answers to the questions posed in the general survey are reported in Table 5. The specific part survey data are presented in the pressure regulator and solenoid valve study, Chapter IX.

c. Alternate Approaches - To eliminate the effect of single failure points and achieve long-life goals, designs that have been developed with quad valves, double seated valves and systems employing valves in redundant configurations.

It doesn't appear feasible that other types of valves can be used in place of check valves. Manual valves are not automatic and would require a crewman to continuously monitor valve position. The sensing controls for using a solenoid valve in this application would be prohibitive.

d. Operating Life - The expected service life of check valves is estimated by the valve manufacturers and users to be 10 years. Taking out the high and low estimates, the check valve cycle life estimates for pneumatic and hydraulic applications varied from 500,000 cycles on low pressure systems to 250,000 cycles on high pressure systems.

The survey response indicated the check valve storage life is determined by age sensitive parts and seats.

Table 5 Results of Manufacturing/Agency Survey - Check Valves

QUESTIONS	CONSENSUS ANSWER
1. Do you manufacture (use) aerospace check valves? Usage?	Yes, the missions are relatively short with the exception of Skylab which requires an eight month life.
2. What is the expected life of subject part?	The cycle life range expected for check valves is 100,000 to 500,000 cycles on up to ten years in duration.
3. What are the failure mechanisms (causes) of the failure modes?	System induced contamination, seat material itself, erosion due to high velocity, joint connections, sliding surfaces.
4. What failure modes would prevent a ten year service life? What failures have occurred?	Internal leakage due to contamination and aging of seals or seat material. Instability of check valve.
5. What solutions do you suggest for the above failure modes that would either enhance the operational life and/or increase the probability of success?	1) Contamination control 2) Contain O-ring seal on three sides 3) Carefully guide poppet on valve seat.
6. To achieve long life, what design features are incorporated in your check valve?	Welded body construction and valve welded into system. Contamination control. Consider redundancy in the valve. Open clearances to allow for non-metal seal swelling.
7. How do you determine part life?	Environmental and qualification testing, cycle tests, and analysis.
8. Did you test for specific failure modes and mechanisms and were any special testing techniques used?	FMEA's were checked against testing to verify there is testing of anticipated failures. Margin testing is performed; however, no particular accelerated life tests are performed.
9. What process controls are necessary to ensure long life?	Most companies indicated a wear-in of 50 to 150 cycles for check valves. Springs and O-rings are both 100% inspected. Ultrasonically clean valves and assemble in clean areas. Units are thermal cycled when directed by the customer.

North American Rockwell (Reference 1) performed a wear analysis on ten selected Saturn S-II valve designs to determine long-life capabilities. They were cycled, disassembled, measured for wear, and recycled. Endurance cycle tests were also performed on selected valves. They concluded that all ten valves were candidates for extended life (several missions). Six of the valves showed promise for long-life (many missions).

The stretchout of the Apollo program caused many components designated for use in the CSM and the Lunar Module adapter to exceed their shelf life prior to use. As a result, a study was undertaken to determine the components with possible age extension. Data from North American report (Reference 1), "Age Life Analysis Sheets" lead to the following conclusions. Plastic seal materials (Teflon, Kel-F, nylon, and mylar) show a 10-year minimum age life. The maximum estimated life expectancy under average exposure conditions was 20 years for Teflon, Kel-F, and mylar (10 years for nylon). Actual test data revealed 9-11 years use with the seals still functioning.

Elastomeric seal materials (silicone rubber, buna-N, Viton, and EPR) show a 10-year minimum age life. The maximum estimated life expectancy under average conditions was 20 years for silicone rubber, Viton and EPR (15 years for buna-N). Actual test data revealed 9-11 years use with the seals still functioning.

Although most manufacturers call out a storage life of three to five years in the procurement specifications, it could be extended as noted by the data above and the actual experience from the Apollo program.

e. *Application Guidelines* - Although interchangeability of the valving elements is possible, the only maintenance or repair of these check valves is complete replacement. Since many of these valves are welded body construction and installed into the system with permanent type connections, such as welded, brazed, or swaged, connections, inflight maintenance is not applicable for these valves.

f. *Additional Studies Required* - Several valve manufacturers state that their valves will operate satisfactorily for 10 years, however, sufficient historical data in a space environment is not yet available to confirm this long life. It is suggested that data be collected to confirm a 10-year service life.

D. TEST METHODOLOGY AND REQUIREMENTS

Check valves are qualification tested according to the customer's specification in the usual manner. The test units are subjected to loads and environment, while monitoring the parameter output for specification conformation. One valve manufacturer has a solenoid valve used as a pilot in a servo valve on their hydraulic test bench and has monitored the life of this valve. During a cycle life of approximately 8,000,000 cycles of 3,000 to 4,000 psi, two problems have been recorded. The tip on the solenoid plunger broke off (a structural failure which was corrected). The second failure not attributable to the solenoid valve; viz, a spring failure on the main shuttle valve. It was thought that check valve springs would be a problem. However, the survey of manufacturers and users reported very few problems attributed to springs.

Although none of those interviewed devised special tests to detect specific failures, failure modes and effects analyses (FMEA's) were used to determine if anticipated failure modes could be detected by test. One response indicated that, according to FMEA, internal leakage was the worst case failure mode for the particular valve. The valve was then tested without the O-ring to determine the leakage rate. It operated satisfactorily. The lives of the valves are estimated from functional and qualification testing or from seals which are subjected to aging.

The only margin testing frequently employed on check valves is pressurization of the valve body to establish safety margins. One manufacturer noted that a margin test or endurance test for cycle life determination was occasionally performed on valves. Thermal cycling was performed on valves only upon customer's request.

It is recommended that contamination tests be performed on critical valves to determine the contamination levels that these components can tolerate. These tests would indicate which components need to be protected with a filter and the required cleanliness level for the system.

When subjected to load, most of the Teflon cold flow occurs during the first 24 hours. For this reason it is recommended that accelerated tests be developed for valves with Teflon seats which require long-life. Initial analysis indicates that if the valve seat is subjected to normal design loads and 150°C temperature for 14 days, the seat would not experience any more appreciable cold flow in a 10 year period.

All valve manufacturers performed 100% inspection of O-ring seals, springs, and wear-in cycle tests on valves, to verify their operation. The wear-in tests on check valves varied from 50-100 cycles and this number of cycles is recommended to eliminate valves which may fail early in their operational life (infant mortality).

1. Apollo Check Valve Test History

The check valve assembly consisted of four, single spring loaded poppet check valves installed in a dual series-parallel arrangement. Differences for check valves for the same propellant system were in poppet seat material and pressure drop.

Initial cracking pressure requirement was 10 psig maximum for the oxidizer valve and 1.5 to 3.0 psig for the fuel valve. Oxidizer valve cracking pressure was reduced from 1.5 to 3.0 psig. Pressure drop requirement for the oxidizer valve was initially 10 psi maximum for 4.5 lb/min. helium flow at 15.6°C temperature and 187 psig inlet pressure. It was reduced and finally fixed at 3.3 psi (3.6 psi for the fuel valve). The internal leakage requirement for the oxidizer valve was initially 25 x 10^{-4} scc/sec. maximum.

Design verification tests were performed during January 1965, with two oxidizer units equipped with Teflon poppet seals. During the high temperature endurance cycling tests one poppet valve stuck open because of a galled and worn poppet stem. The condition was corrected by chrome plating the poppet stems and the test was completed satisfactorily.

Qualification testing for both Block I check valves began in May 1965, and continued to March 1966. Three oxidizer valve units equipped with bonded Resistazine rubber poppet seals and three fuel valve units equipped with bonded Butyl rubber poppet seals were successfully tested. Off-limits testing of two fuel check valves consisting of extended limits vibration and a rupture burst test was completed in January 1966. The units, which were equipped with bonded Butyl rubber poppet seals, met both requirements. The specification requirement was 500 psig minimum. Block I check valves were certified June 1966. The fuel check valve for Block II was qualified by similarity to the Block I fuel check valve. Additional qualification testing required of the oxidizer check valve was performed successfully on two units from April to July 1966. Both check valves were certified in 1968. For subsequent vehicles the following procurement specification design and performance requirement changes were made:

1) Allowable pressure drop was increased;

2) Vibration levels were increased;

3) Filters were installed at check valve inlet and test ports.

The radius of the poppet seal back-up ring was increased to eliminate seal damage. The absolute micron rating requirement for the filter elements installed at the check valve inlets and test ports was not less than 35.0 nor more than 74.0 microns. During filter qualification tests the filter pressure drop was 0.3 psi for 4.5 lb/min. helium flow at room temperature and 186 psig inlet pressure. Specification allowable pressure drop was 1.5 psi. Filter qualification testing was completed in June 1968, with four units successfully meeting all specification requirements.

Qualification testing was performed from May to October 1968, and the check valves were certified February 1969.

E. PROCESS CONTROL REQUIREMENTS

From the survey of valve manufacturers and users it was indicated that the valve parts should receive ultrasonic cleaning and assembly in either clean rooms or on laminar flow benches. It is recommended that the cleanliness requirements be governed by a single contamination control specification that includes the components, fluids and systems. The clean room specification should be such that contaminant size is less than that specified in the valve design specification as being acceptable. Valves should be shipped and stored in bags as a cleanliness control. Consideration should be given to using nylon or polyethylene to prevent creation of contamination due to chaffing by the parts. Vendor controls shall guarantee that the valve contamination particle size and count will not exceed specified limits. Documentation is required on particle levels. The Skylab program required that system cleanliness must extend across interfaces with other contractors.

Customer requirements for critical processes should be called out in the procurement specification. Most manufacturers contacted in the survey stated that plating, heat treat, and rubber products require strict process control.

F. PARTS USAGE CONSTRAINTS

The selection of the ball, cone, poppet, swing, or flapper check valves depends upon the particular application, the limitations, and advantages of each valve type. The TRW Systems Handbook (Reference 2) lists applications and limitations of various check valves. The valve types that appear to offer more advantages for long life include the poppet, cone, and flapper type check valves in the order of decreasing desirability. The use of ball check valves in airborne systems should be avoided due to their tendency to chatter, particularly on rapid closure. Do not use large flat plastic sealing interfaces for long life operation due to high impact wear characteristics. Also don't use rotating butterfly valves for long life applications due to high frictional wear characteristics.

A guideline from the Apollo 13 (Reference 3) incident is to preclude the use of Teflon and other relatively combustible materials in the presence of oxygen and potential ignition sources.

G. REFERENCES AND BIBLIOGRAPHY

PERSONS CONTACTED

1. Mr. Dave Rich, Martin Marietta Aerospace, Denver, Colorado, Department 1640.
2. Mr. Fred Wright, Martin Marietta Aerospace, Denver, Colorado, Department 431.
3. Mr. John Rousseau and Frank Green, Garrett AiResearch, Los Angeles, California, (213) 323-9500.
4. Mr. Bill Day, APCO, Pacoima, California, (213) 896-2411.
5. Mr. Vince Butler, Parker Hannifin Valve Company, Los Angeles, California, (714) 833-3000.
6. Mr. Chuck Alston, J. C. Carter Company, Costa Mesa, California, (714) 548-3421.
7. Mr. F. A. Rusty Dunn, James, Pond & Clark, Inc., Anaheim, California, (213) 625-6951.

REFERENCES

1. *Advanced Technology Studies*, SD71-267, North American Rockwell, December 1971.

2. *Aerospace Fluid Component Designers' Handbook*, RPL-TDR-64-25, March 1967.

3. *Report of Apollo 13 Review Board*, NASA, June 1970.

BIBLIOGRAPHY

1. J. C. DuBuisson, *et al.*; *Handbook of Long-Life Space Vehicle Investigations*, M-70-16, Martin Marietta Aerospace, Denver, Colorado, December 1970.

2. *Advanced Spacecraft Valve Technology Compilation*, NASA Contract NAS7-717, TRW Systems, July 1970.

3. *Study for Cleanliness Level Requirements for Pneumatic and Hydraulic Components - Service Arm Systems, Complex 39*, MCR-69-484, Martin Marietta Aerospace, NASA Contract NAS10-5935, September 1969.

4. R. W. Burrows; *The Long-Life Spacecraft Problem*, Paper No. 67-880, AIAA 4th Annual Meeting and Technology Display, October 23-27, 1967.

5. *Failure Mechanism of Attitude Control System Valve in Chlorine Pentafluoride Service*, Report AFML-TR-71-94, W. D. English and H. D. Samuel, Jr., McDonnell Douglas Astronautics Company, USAF Contract F33615-60-C-1760, July 1971.

6. *Contaminant Particles in Metal-to-Metal Closures*, Report AFRPL-TR-71-112, G. F. Tellier, T. R. Spring, North American Rockwell Corporation, USAF Contract F04611-70-c-0048, December 1971.

7. *Aerospace Mechanisms*, Edited by G. G. Herzl, Lockheed Missiles and Space Company, Distributed by J. W. Stacey, Inc., 1970.

8. Richard J. Salvinski; *Advanced Space Technology for Valves*, ASME Paper 66-MD-61.

9. J. C. DuBuisson; *Long-Life Reliability Problems and Solutions for Valves Under Spatial Environments*, Paper No. VII H.1, Joint National Meeting American Astronautical Society and Operations Research Society, June 17-20, 1969.

10. L. R. Bell, Jr.; *Long-Life Reaction Control System Engines and Valves*, Paper No. 70-603, AIAA 6th Propulsion Joint Specialists Conference, June 15-19, 1970.

11. *Parts, Materials, and Processes Experience Summary*, CR114391, Lockheed Missiles & Space Company, Inc., NASA Ames Research Center, February 1972.

12. *Apollo Service Propulsion System Rocket Engine Bipropellant Valve Improvement Program*, Report 8317-P1, Aerojet Liquid Rocket Company, NASA Contract NAS9-8317, July 1970.

IX. PRESSURE REGULATORS AND SOLENOID VALVES

by P. J. Powell

IX. PRESSURE REGULATORS AND SOLENOID VALVES

A. INTRODUCTION

The long-life problems concerning solenoid valves and pressure regulators in space applications are treated together in this section.

1. Pressure Regulator - General Description

A pressure regulator is a control valve that employs no auxiliary source of power during operation. Regulators control pressure by varying flow as a function of the sensed difference between the actual and the desired pressure. Any unbalanced force resulting from the pressure difference moves a metering element which increases or decreases fluid flow to nullify the pressure error.

Regulators are usually divided into three types:

1) Pressure-reducing regulators reduce the upstream pressure to a predetermined downstream pressure regardless of upstream pressure variations. These regulators are the most common type used in airborne systems;

2) Back-pressure regulators measure and regulate upstream pressure regardless of variation in outlet pressure (also called vacuum regulators), and;

3) Differential pressure regulators maintain a constant pressure across an orifice to maintain a constant flow rate.

A pressure-reducing regulator can be broadly classified as either modulating or nonmodulating. Within each of these two classifications, regulators can be categorized as direct acting or piloted. Other characteristics used to identify regulators include loading method and control action. The basic types of pressure regulators are tabulated below:

Modulating	Nonmodulating
Direct acting, weight loaded, reset action.	Pressure switch and solenoid valve, ON/OFF action (bang-bang).
Direct action, spring loaded, proportional action.	Piloted, spring loaded ON/OFF action.
Direct acting, preset pressure loaded, proportional action.	
Direct acting, constant pressure loaded, proportional plus reset action.	
Piloted, variable pressure loaded, proportional plus reset action.	

Terms associated with pressure regulators are defined as follows:

Modulation - Describes the ability of a regulator to achieve any steady-state flow rate necessary to maintain a constant regulated pressure. Most regulators are designed for modulating flow control. A nonmodulating regulator is simply an ON/OFF device;

Piloting - The use of a small flow control device operated by a small actuation force to control indirectly a large flow requiring a large actuation force;

Loading - The method of achieving a reference force to establish the regulator set point. The reference force is applied to the metering element so that it is opposed by the regulated pressure acting on an attached sensing element;

Regulator Control Action - The relation between a change in metering valve position for a given change in the pressure being controlled. Control action may be two position (ON/OFF), proportional, reset (integral), or proportional plus reset;

Deadband - The variation of regulated pressure about its desired value, usually expressed in percentage of regulated pressure or psi deviation;

Lockup - The condition reached by all pressure regulators when the downstream pressure is increased to the point where the regulator stops all flow. Internal leakage is measured at lockup;

Flow Limiter - Usually a redundant control pilot incorporated into a regulator to prevent the flow rate from exceeding a predetermined value;

Bang-bang Regulator - The most elementary nonmodulating regulator. It is a combination of a pressure switch and a solenoid valve. Although not truly a regulator, this device is often referred to as bang-bang regulator.

A survey of regulator applications and requirements for rocket engine systems is presented in Table 1 to familiarize the reader with general usage. Table 1 and parts of this section were extracted from the Aerospace Fluid Components Designer's Handbook (Reference 1).

The regulator initially used on Minuteman is shown in cross section in Figure 1. The unit is pilot-operated and functions basically as follows. Upstream pressure is admitted through a filter in the inlet fitting to the main flow and pilot poppets, both of which are closed by spring and pressure forces whenever system downstream pressure is equal to regulator lockup pressure. When system downstream pressure drops, the regulator pilot poppet is pushed off its seat via motion of the main reference spring/bellows assembly causing pressure to build up on the underneath side of the auxiliary piston. Sufficient force will eventually be developed, considering the limited capacity of the bleed orifice through the auxiliary piston, to push the main flow poppet off its seat. When system pressure increases adequately, the main reference spring/bellows assembly retracts allowing the pilot poppet to close, pressure equalizes across the auxiliary piston via the bleed orifice, and the main flow poppet closes. Maximum flow is controlled by the orifice assembly built onto the outlet port. Shifts in downstream pressure corresponding to changes in inlet pressure to the regulator are eliminated by the appropriate movement of the flexural member. The net effect is to vary the point at which pilot flow is initiated as a function of closing force developed by inlet pressure on the main flow poppet (e.g., pilot flow starts earlier with increasing inlet pressure).

The main reference bellows was originally an edge-welded assembly; however, the latest regulators incorporate a seamless, formed bellows. Both control poppets are machined from Kynar.

Table 1 Pressure Regulators

System	Application	Regulating Stages	Set Point (psig)	Flowrate (lb_m/sec)	Orifice Size (in.^{2})	Accuracy	Inlet Pressure Maximum (psig)	Inlet Pressure Minimum (psig)	Inlet Temperature Maximum (°F)	Inlet Temperature Minimum (°F)	Ambient Temperature Maximum (°F)	Ambient Temperature Minimum (°F)	Minimum Ullage (ft^3)	Fluid	Sensing
Jupiter	Guidance	--	28 to 45	0.024	0.0086	±0.2	3,000	200	125	-65	100	85	Very Small	Nitrogen	Internal
Jupiter	Roll Control	--	350	0.5	0.052	5.0	3,000	500	165	-65	165	-55	Very Small	Nitrogen	Internal
Thor	Fuel Tank Pressurization	--	25	0.07	0.026	1.5	3,500	700	160	-100	160	-65	5	Helium	Remote by Probe Entering through Outlet Line
Atlas	Fuel Tank Pressurization	--	59.9	0.6	--	1.5	3,000	75	450	-100	125	-65	8	Helium	External Using a 1/4-in. Sense Line
Atlas	Oxidizer Tank Pressurization	--	24.7	0.6	--	0.6	3,000	75	450	-100	125	-65	--	--	External with a 1/4-in. Sense Line
Titan I	First Stage Fuel Tank Pressurization	--	12.0	0.1	0.045	0.5	3,250	200	290	-250	160	-35	25	Helium	External
Titan I	First Stage Oxidizer Tank Pressurization	--	34.0	0.11	--	0.5	3,250	200	290	-260	160	-35	--	Helium	--
Saturn S-IVB	Fuel Tank Pressurization	--	250	0.18	--	±25	3,100	400	160	-423	160	-65	--	Helium	Internal
Apollo Service Module	Reaction Control System	--	173	0.13 0.20	--	2	4,500	250	150	-65	160	-65	2	Helium	--
Apollo Command Module	Reaction Control System (Primary) (Secondary)	Series Redundant	291 295	0.0050	0.0906	±4.0	4,500	400	110	-65	150	+30	0.058	Helium	Internal
Saturn S-IVB	Aux. Power System Tank Pressurization (Primary) (Secondary)	Series Redundant	196 200	0.0033	0.0017	3.0	3,200	350	110	-65	160	-30	0.116	Helium	Internal
Nimbus	Attitude Control	Single	35	0-0.0063	0.12	2.0	3,900	200	200	-60	200	-60	0.0012	Nitrogen	Internal
	Attitude Control	Series Redundant	380	5.1	0.35	±20	4,800	500	170	-100	170	-60	0.021	Freon No. 4	Internal
Surveyor	Fuel and Oxidizer Pressurization	Single	725	0.009	0.07	20	5,200	820	115	-150	115	-20	0.04	Helium	Internal
	Fuel Cell	Single	1.0	6.13x10[illegible]	0.008	±0.2	30,000	25	160	-1.0	160	-65	0.0001	Hydrogen	Internal
Mercury Gemini Apollo	Life Support	Single	80	0.004	0.08	±10	7,500	200	200	-80	200	-50	0.001	Oxygen	Internal
NASA Reentry	Electronics Pressurization	Single	1.50	0.00061	0.027	0.075	5,000	200	165	-80	165	-60	0.001	Helium	Internal
Apollo	O Regulator	Single	3.45	0.0007	0.035	2	7,500	200	200	-80	200	-50	0.001	Oxygen	Internal
Ranger	Cold Gas Attitude Control	Single	15	0.0015	--	±1.0	4,000	150	160	-10	160	-10	0.0029	Nitrogen	Internal
Mariner	Cold Gas Attitude Control	Single	15	0.0015	--	1.0	4,000	150	160	-10	160	-10	0.002	Nitrogen	Internal
OGO	Cold Gas Attitude Control	Single	50	0.003	--	±1.5	4,000	150	140	-10	140	-10	0.0029	Nitrogen	Internal
Lunar Orbiter	Cold Gas Attitude Control	Single	19.5	0.0027	--	1.0	4,000	150	160	-15	160	-15	0.0029	Nitrogen	Internal
Pioneer	Cold Gas Attitude Control	Single	50	0.003	--	±1.5	4,000	150	160	-15	160	-15	0.0012	Nitrogen	Internal
Vela	Cold Gas Attitude Control	Single Leakage Redundant	50	0.0022	--	±2.5	4,000	200	140	-10	140	-10	0.0014	Nitrogen	Internal
Bio-Satellite	Cold Gas Attitude Control	Single	35	0.003	--	±1.5	4,000	150	160	-20	160	-20	0.0029	Nitrogen	Internal
Lunar Orbiter	Velocity Control Engine, Propellant Tank Pressurization	Single	180-200	0-0.0071	--	1.0	3,820	650	125	-65	85	35	--	Nitrogen	Internal

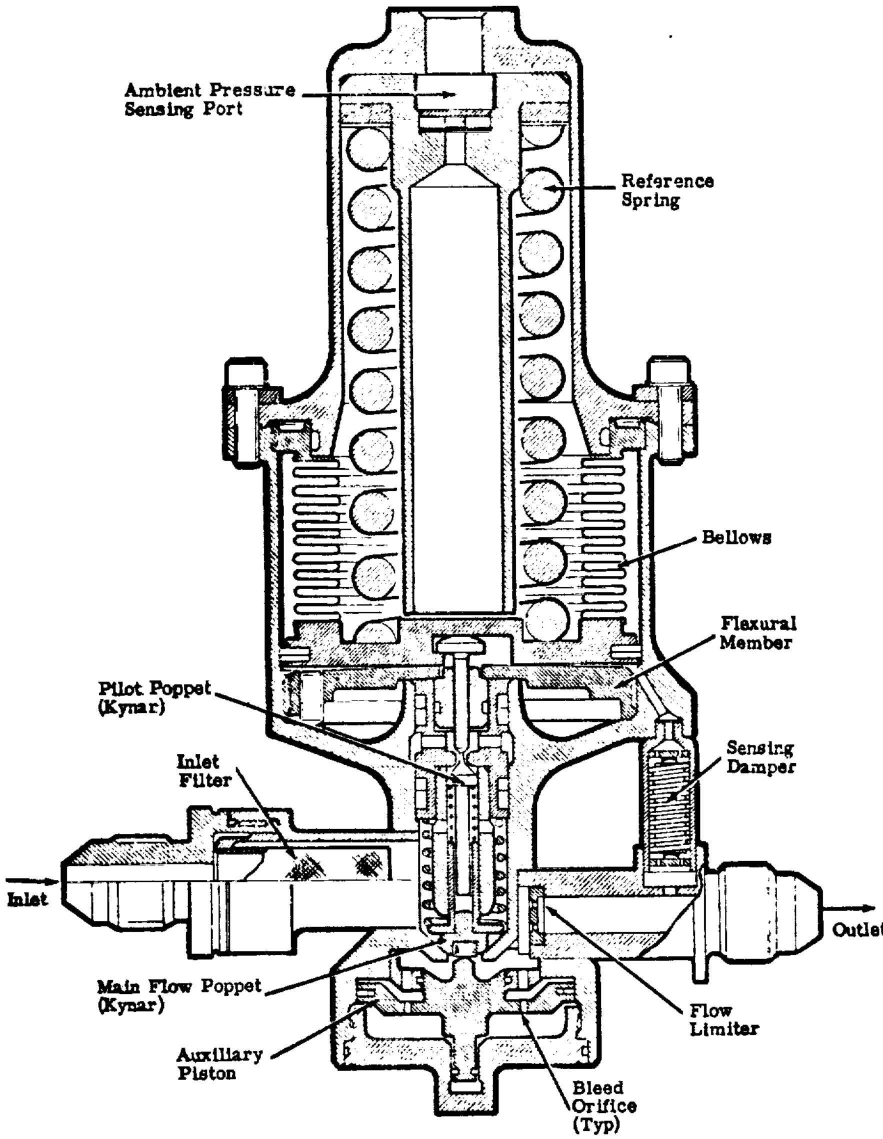


Figure 1 Cross Section of the Minuteman Pressure Regulator

The pressure regulator currently used on the Minuteman missile (Figure 2) and Mars Mariner has the following features:

1) A single plunger is used between the bellows and the poppet. The result is less friction and a smaller package size and weight;

2) There are no moving parts in the flow limiter. The Minuteman unit utilizes a fixed venturi built into the inlet fitting to control maximum flow;

3) The use of a single bellows, formed from seamless tubing, results in an 11-psi outlet pressure control band due to the low spring rate involved. A higher unit reliability results from a seamless bellows compared to a redundant edge-welded bellows (Fig. 3), and;

4) A total of 10 Minuteman regulators were successfully qualification tested without significant failures.

The LM pressure regulator assembly (Fig. 4) consists of two identical pressure regulators in series. Each regulator maintains its own outlet pressure. The predominant failure mode of each regulator is "fail open." Thus, the outlet pressure remains within the control band after the occurrence of a single "fail open" malfunction. During normal operation, there will be only a very small ΔP across the downstream regulator.

The valve is positioned by the force balance between a pressure sensing bellows and a Belleville spring assembly. If regulated pressure falls, the sensing element opens the valve and admits more gas, tending to maintain a constant pressure. Accurate pressure regulation is achieved by utilizing a sensing element with a large effective area and a low spring constant.

The inlet pressure exerts a force on the valve seat. This force is a part of the force balance that determines regulated pressure, and if not compensated, would cause undesired variations in regulated pressure. "Pressure balanced" inlet valves compensate for this effect, but require high pressure sliding seals which lower reliability. In the LM regulator, compensation is obtained by mounting the valve seat on a stiff but compliant plate, which deflects slightly with inlet pressure.

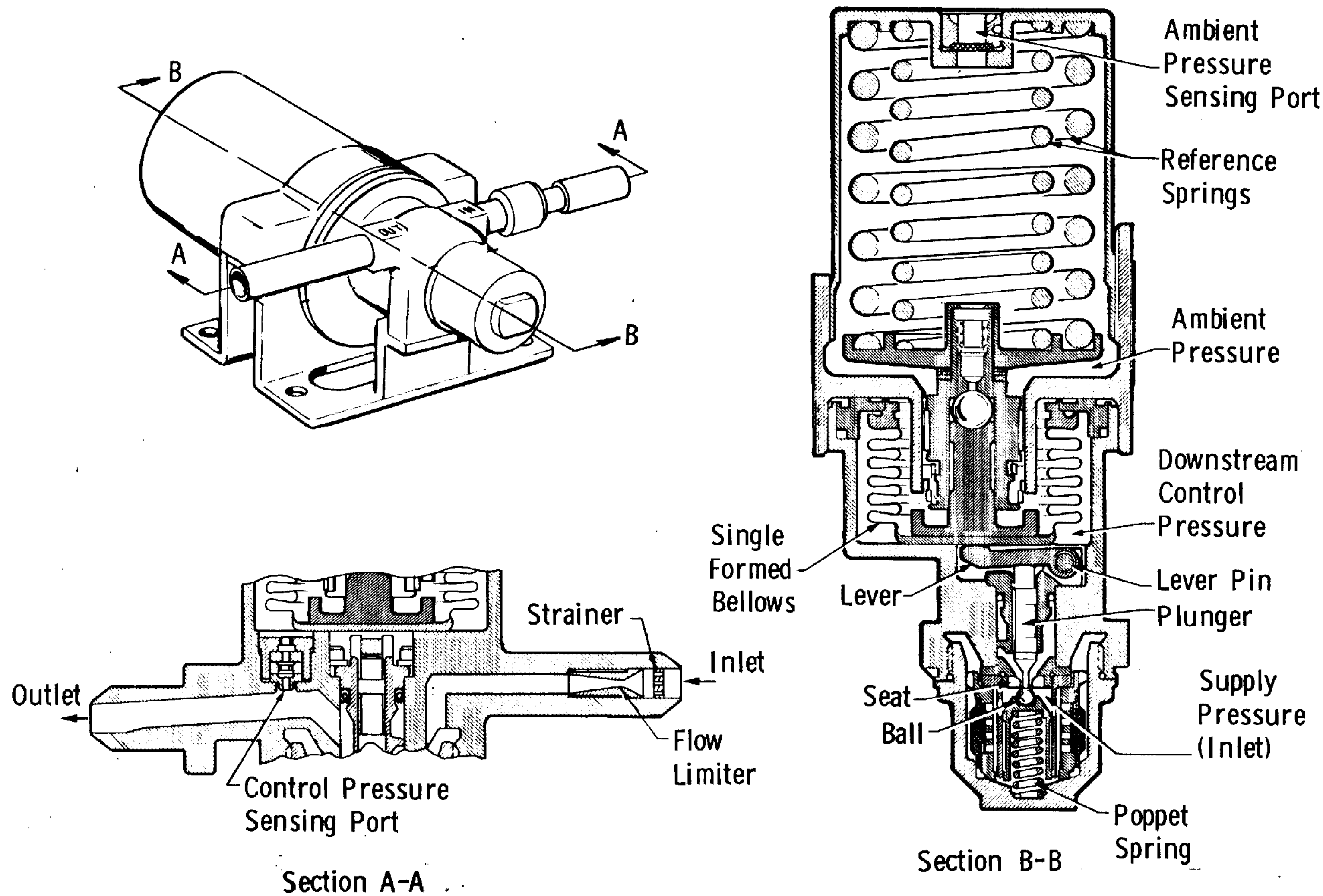


Figure 2 Pressure Regulator

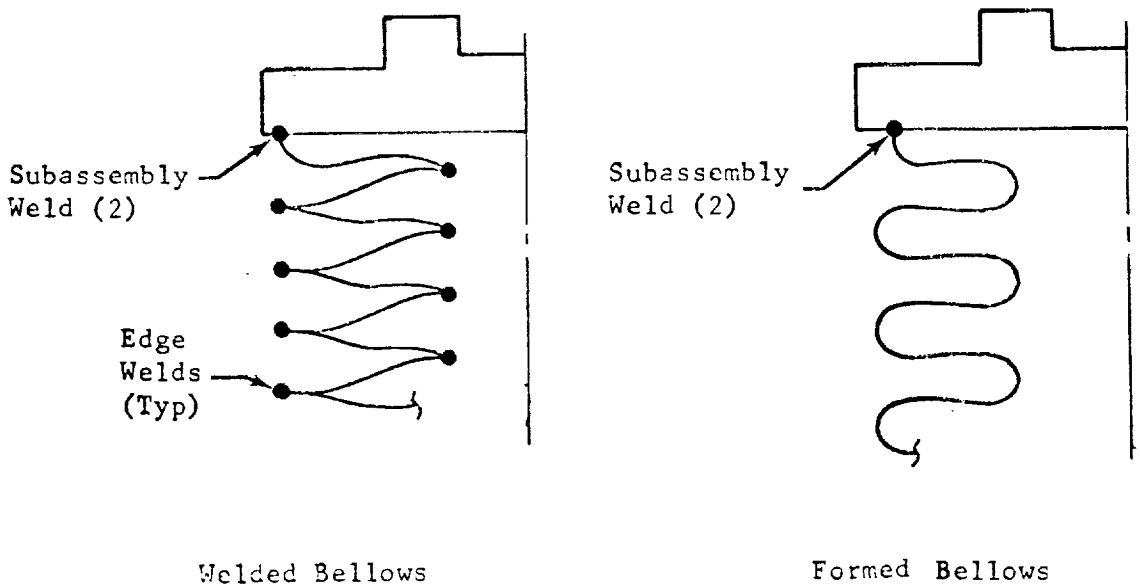


Figure 3 Cross Section of Edge-Welded Bellows Compared to Formed Bellows

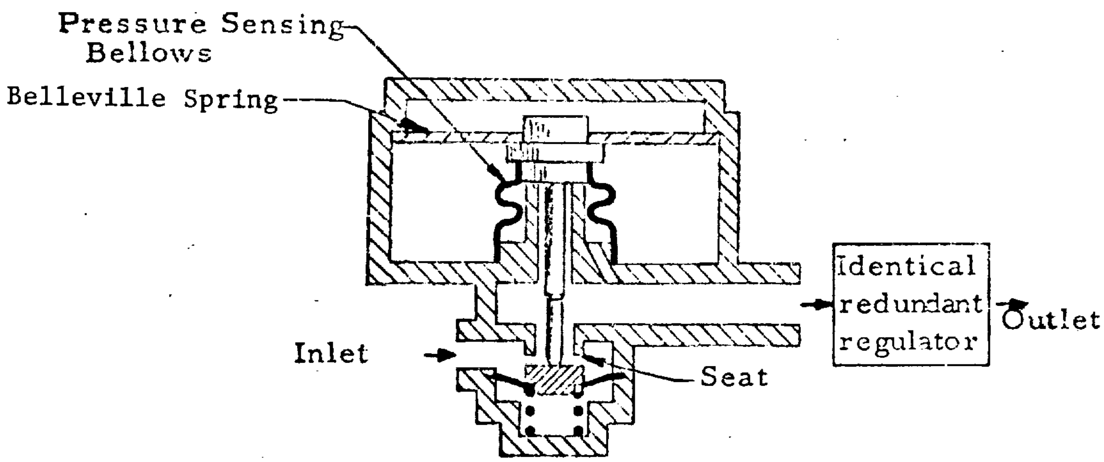


Figure 4 Schematic Diagram of the LM Pressure Regulator

2. Solenoid Valves - General Description

Virtually any type of valve can be solenoid operated. A solenoid valve consists of two major functional assemblies: an electromagnetic assembly consisting of an electromagnet and an electromagnetic plunger in which a poppet is affixed, and a valve body assembly containing an orifice and means to attach external lines and hardware. Solenoid valves are normally classified as shutoff or latching.

Valves are designed in normally open or closed positions. In the normally closed design, the valve opens when the current is applied to the solenoid, and closes when the current is removed. The normally open valve is just the reverse; the valve closes when the current is applied, and opens when the current is removed. Figure 5 shows a typical normally open solenoid shutoff valve. A spring is utilized to maintain the poppet in the fully open position. When current is applied to the electromagnet, the movable core plunger travels into the housing forcing the poppet into the seat to prevent flow through the orifice. The spring is now in compression, and the spring force will return the movable core plunger to the open position when the current is removed from the electromagnet.

A latching solenoid valve used on Apollo in the supercritical hydrogen and oxygen gas storage systems is depicted in Figure 6. The valve uses two solenoids on opposite sides of a common armature for valve actuation and a Belleville washer for the latching function.

To open the valve, solenoid 1 is energized to provide a pull force on the armature which in turn provides an opening force to the poppet. When the armature has moved a sufficient amount, the Belleville washer snaps and holds the poppet off the seat. Solenoid 1 is now de-energized and the valve remains open.

The valve is closed by energizing solenoid 2. This pulls the armature down and snaps the Belleville washer to the opposite mode. The pressure difference across the poppet and a small coil spring force hold the valve closed. Solenoid 2 is now de-energized and the valve remains closed. The position switch obtains its position indication from the armature and supplies a closed circuit when the valve is in the closed position.

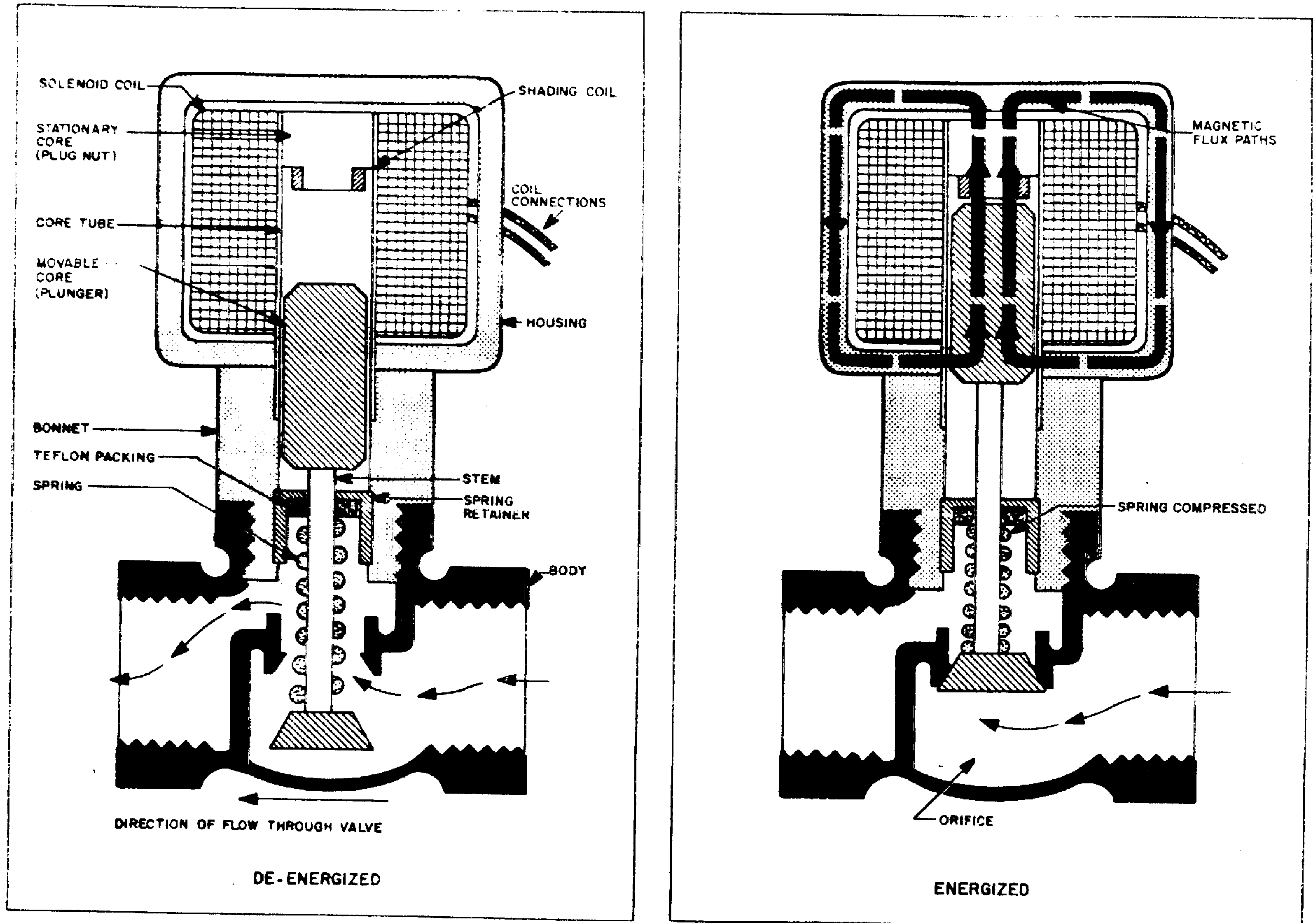


Figure 5 Direct Acting Solenoid Valve

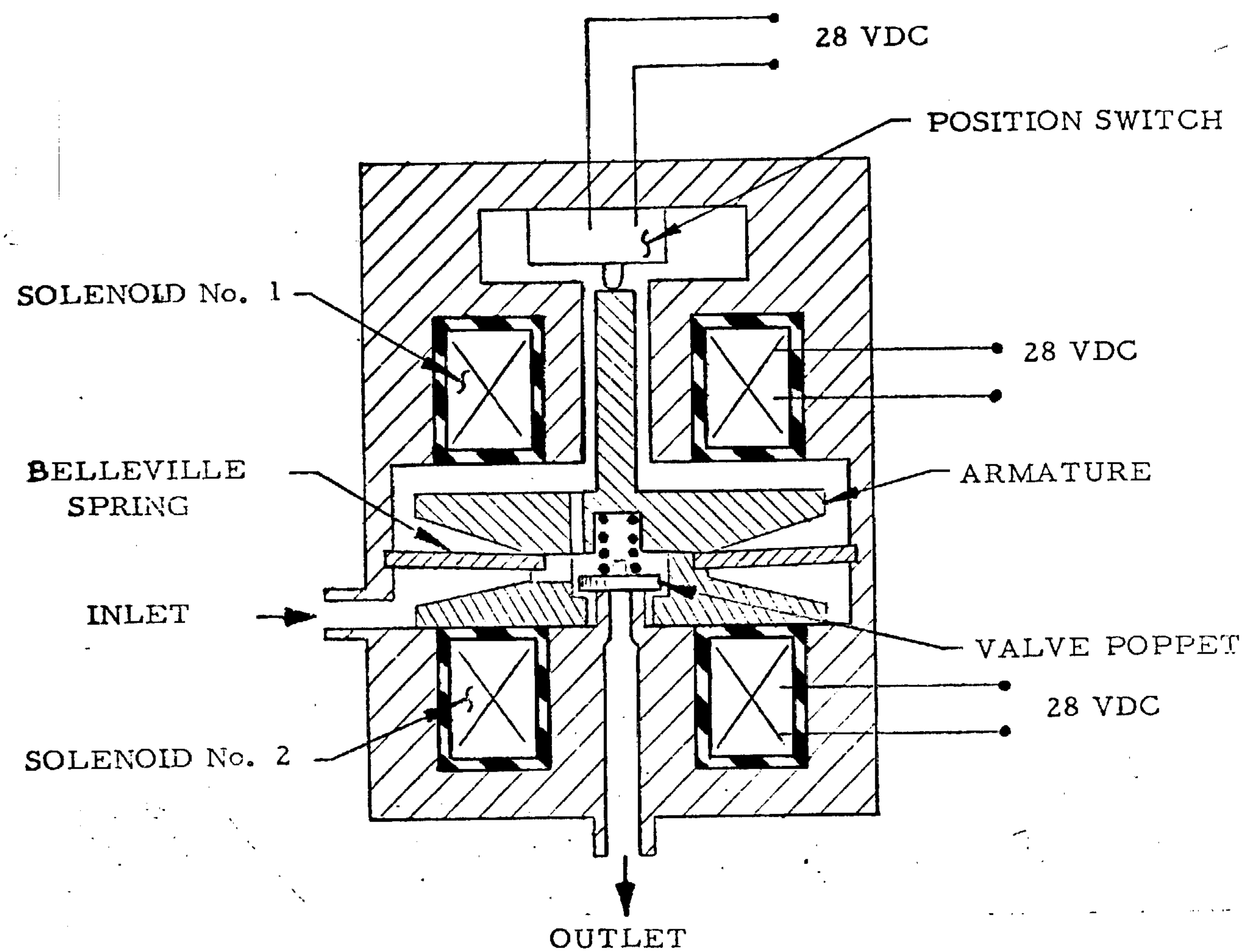


Figure 6 Schematic Diagram of a Latching Solenoid Valve

When valves are specified as two-way, three-way, etc. the digit always refers to the number of ports--the openings or passage ways through which the gas or liquid (medium) flows. Two-way valves must be specified as normally open or closed. If the valve operation uses more than two ports (three-way, four-way), it must be described in words or by a schematic to indicate which ports are interconnected when the solenoid(s) are energized or deenergized.

Pilot operation is useful when insufficient power is available to actuate a large valve. A smaller "pilot valve" is first actuated admitting fluid to apply pressure for actuation of the larger valve.

B. GUIDELINES FOR LONG-LIFE ASSURANCE

The current state-of-the-art solenoid valves and pressure regulators can meet the anticipated requirements for long-life applications such as the Shuttle program. Both spring return solenoid valves and pressure reducing regulators can currently obtain calendar service lives of 10 years without maintenance. For gas applications, a solenoid valve cycle life of 500,000 cycles is currently possible; and 1,000,000 cycle life is obtainable with an advanced state-of-the-art valve. A pressure regulator cycle life of 500,000 cycles is currently possible; a cycle life of 1,000,000 is obtainable for environmental control system (ECS) applications.

Internal leakage caused by contamination is the major life-limiting failure mode for the valves studied. The contamination may be from either the fluid medium or from valve wear particles. Listed in the order of most probable occurrence, the solenoid valve failure modes are:

1) Internal Leakage;

2) Failure to open or close, and;

3) External Leakage.

Listed in the order of most probable occurrence, the pressure regulator failure modes are:

1) Internal Leakage;

2) Regulator Fails Open or Regulates High;

3) Regulator Fails Closed or Regulates Low, and;

4) External Leakage.

1. Design Guidelines

1) Design large flow paths (areas) through poppet solenoid valves and main sealing surfaces to reduce flow velocities and erosion of seats.

2) Configure housings (as far as possible) to eliminate areas that can entrap contaminants. Use a "swept by" design where flow will tend to pass contaminants through the component.

3) Insure that materials are compatible with the working fluid including maximum expected fluid impurities. The evaluation must consider dynamic as well as static applications. It must also consider temperature, pressure, and phase variations of the fluid.

4) Minimize component induced contamination:

   a) Material selection should consider the effect of wear particle size on useful life. Particle sizes vary as The Young's Modulus divided by the square of the compressive yield stress.

   b) Use rolled threads in preference to machined threads to minimize burrs and achieve higher strength.

   c) Minimize dead-end passages and capillary size passages.

5) Leaks due to contamination are minimized if the seat loads are sufficient to plastically yield a trapped contaminant particle on the sealing surface. System contaminants must be identified to determine particle size and material properties.

6) Avoid sliding parts; they not only induce contamination, but entrap contamination.

7) Develop valving elements with quick opening areas to preclude chattering, instability and high seat velocities.

8) Spherical or tapered poppets are recommended. These poppets are self-aligning, but require precise alignment of the seat and the poppet assembly.

9) Insure guiding of poppet onto seat to allow for maximum alignment and eccentricity tolerance (i.e., use a large length/diameter bearing surface to guide ratio - minimum 2:1).

10) Justify not employing elastomer seats. Low leakage rate requirements of less than 5 scc/hr can be most easily met by elastomer seats.

11) Since low seat stresses are desirable for long life, high safety factors should be used in conjunction with high seat loads.

12) Control the undesirable creep or cold flow characteristics of Teflon seats by:

    a) Containment of Teflon seats on three sides;

    b) Heat stabilization techniques, and;

    c) Low stress levels.

13) Avoid Teflon seals for high pressure oxygen system applications to preclude flammability hazards.

14) Seat stress of plastic seats must be held well below material yield stress for long-life storage to prevent excessive deformation. Stresses in elastomeric seals are generally not critical.

15) Wear is minimized by making the hard seat in the housing wider than the plastic seal of the poppet or conversely by making the metal sealing surface of the poppet wider than the plastic seat.

16) Smooth surface finishes are superior for long life; therefore, it is recommended a finish of 16 - 32 rms be used.

17) Lipseals supported with a back-up ring to resist undesirable influences of pressure loading are superior to designs without this provision. It is recommended that lipseals guiding piston rods or poppets use back-up sealing rings in addition to the main lipseal.

18) Cold-formed plastic lipseals and certain machined plastic lipseals that rely on diametrical stretch to effect a seal such as rotating butterfly valves should not be used for long life applications due to the high frictional wear characteristics.

19) Avoid plastically deformed soft metals. This configuration has excellent sealing characteristics, but exhibits low life.

20) Do not use large flat plastic poppet sealing interfaces. They are not compatible with long-life due to high impact wear characteristics and misalignment.

21) Elastomeric seals may be contained by mechanical or bonding methods. Use bonded seals only where proven in application and use 100% nondestructive test inspections.

22) To minimize transient stresses due to poppet travel, provide positive stops at the end of travel.

23) Seal retention methods shall prevent seal distortion, creep or dislodgement.

24) The inherently larger leakage rates of hard or hard seat configurations can be minimized by lapping poppet and seat to obtain better seat finishes.

25) Reduced life due to vibration sensitivity is minimized by decreasing clearances in bearings and guides, avoiding large overhung moments, and restraining lateral motion of poppets.

26) Provide external fittings to permit drain, flush and purge after exposure to propellants.

27) Control stress corrosion by avoiding stress corrosion susceptible materials or design parts to operate at low stress levels.

28) Justify the use of dynamic seals. They are subject to wear and cause unpredictable drag.

29) Control external leakage by:

    a) Requiring welded valve body construction;

    b) Require vacuum melt bar stock, and;

    c) Impregnate valve casting with sealant.

30) Avoid tapered plug valves and gate valves since they are susceptible to sticking.

31) The solenoid open and close force margins goal is 300%. Failure of the poppet to open (or close) may be caused by binding of the plunger or insufficient solenoid force.

32) Prevent insulation deterioration and subsequent solenoid shorts that can lead to reduction in electromagnetic force by:

    a) Coating solenoid valve lead wires with an abrasive resistance covering;

    b) Take special precautions when joining the lead wire to the solenoid coil wire, and;

    c) Pot coils and lead wires to prevent movement during vibration and operation.

2. <u>Process and Control Guidelines</u>

1) Ultrasonically clean valve parts; assemble in specified level-clean areas, and govern by a single contamination control specification during test. This same control specification should also govern the test fluid media used.

2) Use a fabrication barrier (bag) to protect clean parts. Consider using nylon or polyethylene to prevent creation of contamination due to chaffing of the barrier by the parts.

3) Vendor controls shall guarantee that the valve contamination particle size and count will not exceed specified limits. Documentation is required.

3. Test Guidelines

1) Conduct contamination susceptibility tests during development to determine the level of contaminant that the valve can tolerate.

2) To eliminate components with latent manufacturing defects, conduct 50 to 100 run-in cycles.

3) Do not rapid cycle valves (designed for fluid applications) for functional verification in a dry condition because the lack of fluid damping can increase seat stress and reduce life.

4) Conduct life cycle endurance tests under operational conditions. For long-life applications, the valve cycle life parameter must be known.

5) Require calibration logs be kept on pressure regulator assemblies to determine if the unit is drifting out of limits prior to service usage.

6) Test for corrosion of materials in the presence of the working fluid and the maximum expected impurities expected during operational life. Evaluate pitting and stress corrosion as well as penetration rates over a large surface area.

7) Consider use of holographic terferometry test methods for fluid compatibility evaluations. These methods allow evaluation of the onset and time variation of the corrosion process and permit three dimensional evaluation of localized effects.

8) After operational valve use, measure the response characteristics of the valve and perform trend analysis to identify wear trends.

9) Consider means to minimize cold flow during service. Initial analysis of an accelerated test for Teflon valve seat/poppet assemblies indicate that if the assembly is subjected to normal design loads and 150° C temperature for 14 days, then the assembly would not experience any more appreciable cold flow in a ten year period.

4. Application Guidelines

1) To avoid contamination and to prevent leakage, install components into systems using permanent connections, such as welded, brazed, or swaged connections rather than reconnectable mechanical joints.

2) Solenoid valves and pressure regulators are not normally repairable by in-flight maintenance procedures.

3) System design should identify the Line Replaceable Units (LRUs) consisting of individual valves or regulators, or groups of valves or regulators, for ease of remove/replace activities in the event of component malfunction.

4) A solution to reliability problems is the application of valves and regulators into redundant configurations.

C. LIFE LIMITING PROBLEMS AND SOLUTIONS

1. Failure Mechanisms Analysis (FMA)

The failure modes (Table 2) that could materially affect the functional reliability of solenoid valves will be individually discussed in the following paragraphs. The failure modes listed in their most probable order of frequency are:

1) Internal leakage (through the valve);

2) Failure to open or close, and;

3) External valve leakage.

The failure modes (Table 3) that could materially affect the reliability of a pressure regulator valve are:

1) Internal leakage (through the valve);

2) Regulator fails open or regulates high;

3) Regulator fails closed or regulates low, and;

4) External leakage.

Except for the failure modes associated with the source of valve actuation, the failure modes of solenoid and pressure regulator valves are essentially the same. Internal and external leakage failure modes for both valve types are discussed together.

a. Internal Leakage - Internal leakage through a valve is the most serious of the functional problems for long duration missions because (1) even a small leakage rate can deplete the supply of the flowing medium,* and (2) if the flowing medium is corrosive or explosive, damage to equipment or personnel can result. Most of the literature and valve authorities surveyed agreed that soft seats should be employed to obtain the very low leakage rates required for space applications. Because of their relative resiliency, soft seats provide better leakage control than hard seats;

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*Zero leakage is defined by JPL as any leakage equal to or less than 10^{-7} scc of helium per second.

Table 2 Failure Mechanism Analysis - Solenoid Valves

Part and Function	Failure Mode	Rel. Rank	Failure Mechanism	Detection Method	How to Eliminate/ Minimize Failure Mode
Valve seat - (seal flow media)	Internal leakage	1	Internal leakage is caused by: 1) Contamination 2) Damaged seat 3) Seat wear	<u>Pre-Installation</u> Testing and valve run-in will detect this type of failure. <u>Post-Installation</u> System pressure measurements can be used to determine if valve is leaking.	Control cleanliness level for parts, components and systems. Areas that can trap contaminants should be eliminated from valve designs (use a "swept by" design where flow path is through the valve).
Poppet assembly - (control flow)	Failure to open or close	2	Poppet stuck in intermediate stroke position due to: 1) Contamination 2) Misalignment of poppet 3) Solenoid failure	<u>Pre-Installation</u> Testing will determine this type of failure prior to valve installation into a system. <u>Post-Installation</u> Valve position indicator or system pressure measurements.	Allow conservative force margins for opening and closing of valve seat. Run-in test or margin tests should reveal this type of failure.
Valve body - (support valving elements and contain media)	External leakage.	3	Leakage through: 1) Static valve seals 2) Plumbing connections 3) Valve body (porosity)	<u>Pre-Installation</u> Component tests will reveal this failure mode. <u>Post-Installation</u> System pressure (fluid) loss or visual detection of leaks for manned flight.	Methods to control these failures include: 1) Welded external body construction 2) Install valve into system with permanent mechanical connections 3) Impregnate castings with sealants 4) Use of vacuum melt metals to control inclusions or stringers.

Table 3 Failure Mechanism Analysis - Pressure Regulator

Part and Function	Failure Mode	Rel. Rank	Failure Mechanism	Detection Method	How to Eliminate/ Minimize Failure Mode
Poppet/seat assembly - (seal flow media)	Internal leakage	1	Internal leakage is caused by: 1) Contamination 2) Damaged seats 3) Seat wear	<u>Pre-Installation</u> Part inspection and run-in tests. <u>Post-Installation</u> Audible or by pressure sensor.	Provide cleanliness control for parts, components, and systems. Use design features which control plastic seat cold flow.
Sensing and actuation element - (control pressure)	Regulator fails open or regulates high	2	Contamination between moving parts can slow the actuation mechanism resulting in loss of pressure (possibly out a relief valve).	<u>Pre-Installation</u> Run-in tests and log of calibration. <u>Post-Installation</u> Failure is detected by on-board displays or pressure downstream of the regulator.	Provide cleanliness control for parts, components, and systems. Areas that can trap contaminants should be eliminated from regulator design. Install a filter upstream of pressure regulator.
	Regulator fails closed or regulates low	3	Spring creep or relaxation resulting in loss of pressure regulation path.		Provide 100% inspection of parts, and provide run-in test to measure if the sensing element has a tendency to drift out of limits.
Regulator housing - (support actuation element and contain flow media)	External leakage	4	Leakage through: 1) Static seals 2) Plumbing connections 3) Regulator housing (porosity)	<u>Pre-Installation</u> Component pressure tests reveal this type of failure mode. <u>Post-Installation</u> System pressure loss or visual detection for manned flights.	Methods to control these failures include: 1) Welded housing construction 2) Install regulator package into the system with permanent mechanical connections 3) Impregnate castings with sealant 4) Use of vacuum melt metals to control inclusions or stringer.

however, the limitations of propellant compatibility, thermal compatibility, adverse effects on valve time, and repeatability often dictate the use of hard seats. Metal-to-metal seals can maintain the low leakage rates required if very careful design and quality control are employed; however, the probability of a successful metal-to-metal seal is less than that for a metal-to-elastomer seal. In addition, the probability of poppet-to-seat cold weld is greater for metal-to-metal seals than metal-to-elastomer seals which tend to vulcanize when closed for long periods.

Teflon or Kel-F seats are in common usage where low leakage rates are required after repeated cycling. Kel-F is used when the pressure differential is greater than 200 psi. Teflon seat stresses should not exceed 3000 psi to avoid cold flow and subsequent leakage. Teflon seats should not be used above 150° C because the seats are easily damaged by large particles and vibration at higher temperatures. Cold flow of soft seat has been reported in numerous applications. A number of good design practices have been developed at the various NASA centers to solve the problem. The majority of the solutions consist of surrounding the soft seat material on three sides with a metal backup and then closing the poppet against the fourth surface.

At the National Bureau of Standards (Ref. 2), Washington, D. C., techniques for fabricating improved flange seals for vacuum as high as 10^{-8} torr have been developed by Dr. R. L. Anderson, Dr. L. A. Guildner and R. E. Edsinger. These seals, fabricated from TFE gaskets, are pressurized beyond the yield point by spring-loading with a flexed metal bar. This pressure is maintained while the seal is in use, and flow of the material is confined to the sealing area. To avoid excessive flow, clearances between parts are held to 0.025 mm maximum.

Figures 7 and 8 show two types of flange seal applications. The yield point for TFE is 3000 psi and when subjected to greater pressure, the material exhibits flow that is mainly elastic. As a result, it will seal surfaces of relatively poor quality. Unstressed, the TFE is somewhat porous, but under pressure that exceeds the yield point, it becomes impervious.

Contamination is the major failure mechanism or functional problem. Contamination can cause valve leakage, sticking of sliding surfaces, increased wear, plugging of small orifices, scoring, and high friction forces. According to Ref. 3, failures attributable to contamination accounted for 10 of the 41 Titan failures.

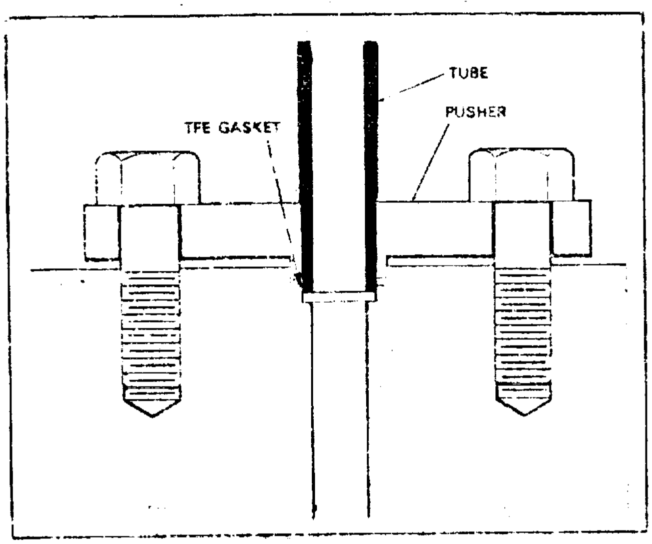

Figure 7 *Cross section of gasket is made small to minimize sealing force and possibility of permeation. Shoulder stop for end of tube helps provide alignment and indicates that tube has been inserted past gasket.*

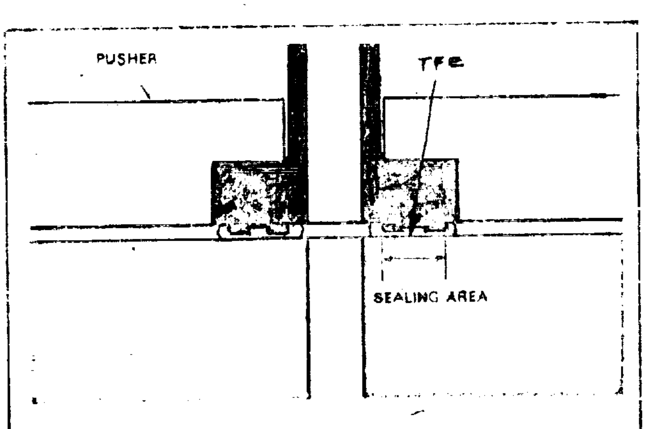

Figure 8 *TFE seal between two flat surfaces would fail because of porosity and excessive flow of TFE under pressure. Land-and-groove seal copes with this problem. As lands squeeze into gasket, TFE is forced into groove under pressure. For 0.25-mm-thick gasket, groove would be 0.15 mm deep.*

In addition, a Milmanco Corporation of Seattle, Washington, study for USAF of missile valve failures found 21 failures due to contamination. Experience at Martin Marietta's Denver Division indicates that the source of most contamination has been the vendor facilities. The major cause of the problem has been a lack of definition and specification of contamination and its limiting controls. It is recommended that:

1) Components be ultra sonically cleaned and assembled under specified level clean area conditions;

2) Areas that can entrap contaminants should be eliminated from the design;

3) Written procedures should be implemented and enforced to assure cleanliness not only of the component, but of the system to which it is installed, and;

4) Filters be placed upstream of critical components, such as pressure regulators.

Extensive tests on 16 separate filters by Space Technology Laboratories revealed a universal lack of initial cleanliness of the filters in the as-received condition. Almost all the filters contained built in dirt or contamination particles that were in excess of the absolute rating of the filter. It is recommended that strict controls be implemented to ensure the cleanliness of filters.

For long duration missions, contamination could be caused within the system by wear and chemical action of the medium (especially propellants) on the valve materials. Only very careful design and material selection can reduce the probability of this source of contamination to acceptable levels. The valve seats themselves will produce wear particles; their size is a function of the material and fluid environment. Filters cannot protect the moving parts from wear particles created by the valve parts.

b. *External Valve Leakage* - External leakage is caused by leakage through or around static seals. To eliminate this failure mode, welded body construction is preferred and permanent connections such as brazed, welded, or swaged should be used when the components are installed into the system. More details of these failure modes (including valve housing porosity) are presented in Chapter VIII.

c. Solenoid Valve Failure to Open or Close - Failure of a solenoid valve to stop, allow, or control the flow rate can be caused by sticking or sliding surfaces, insufficient electromagnetic force, insufficient return spring force, or perhaps freezing of the fluid medium.

Sticking (high friction forces) of sliding or contacting surfaces can be caused by (1) cold welding, (2) inadequate lubrication, (3) contamination, or (4) incorrect design. Contamination has been discussed in previous paragraphs.

Excessive pressure drop is caused primarily by either contaminant reducing the valve seat orifice area or by failure of the poppet to fully withdraw from the seat. In aerospace applications where great care is taken to ensure clean systems and efficient filtering, the probability of excessive pressure drop due to the collection of contamination is deemed negligible. Flow through the valve should remove most of the contaminant. Employing the largest orifice possible reduces the probability of excessive pressure drop. Failure of the poppet to fully open may be caused by binding of the plunger and/or insufficient solenoid force to fully actuate the plunger.

Insufficient electromagnetic force may occur if the frictional forces discussed previously are excessive or the solenoid coil fails open or shorted. Thermal cycling can cause the solenoid circuit to either open or short due to repeated expansions and contractions. The lack of thermal convection in a vacuum must be considered in the design. The solenoid insulation must be compatible with the fluid medium and the radiation environment to prevent insulation deterioration and subsequent solenoid short(s) that can cause either a complete or a partial reduction of the electromagnetic force. To enhance reliability it is suggested that:

1) All wires and coils be potted with a material that has good thermal conductivity, but which imposes minimum stress on the encapsulated wiring;

2) Open and closure force margins be conservative (a margin of 300% is suggested). Verify these margins by test;

3) Redundant solenoids be considered;

4) Use wire gages less than 40 gage;

5) Coat lead wires with abrasive resistance covering such as Kapton, H-film, or polymide;

6) Take special precaution when joining the lead wire to the coil wire;

7) Avoid sliding parts.

Insufficient return spring force may be caused by binding or loss of spring force due to annealing from thermal cycling or radiation. The probability of binding can be reduced by providing the proper clearances even with thermal and load distortions. Marshall Space Flight Center (MSFC) had a problem involving springs used in position indicators during thermal cycling tests. The initial hardness of a spring may have been obtained through heat treatment; an annealing effect will occur with temperature cycling that can cause springs to take a permanent set. In addition, temperature cycling of valves will relieve stresses left in the valve during their manufacture. This results in warpage and out-of-tolerance parts. Sliding parts may bind and fail to operate because the spring or solenoid has insufficient force to operate the parts. Although the probability is low because of shielding, solar radiation can produce high thermal stresses and should be considered in the design. The following suggestions should reduce the probability of insufficient return spring force:

1) Use Inconel-X for spring material in cryogenic application;

2) Consider using the ball-poppet in applications where thermal expansion, warpage, or annealing of built-in stresses may be a problem. Geometrically, a sphere is less susceptible than other shapes to physical distortion due to temperature changes. Hollow spheres exhibit dimensional stability superior to solid spheres. Ceramic spheres used for low temperature applications had a wear rate five times less than that of CRES spheres, but are not always compatible with the flow medium.

3) Hard drawn steel or music wire is not recommended as spring material.

Expansion through the valve orifice can cause freezing of the fluid medium. There is some indication that a Titan IIIC transtage bipropellant valve may have failed due to frosting. It is suggested that:

1) Thermodynamic calculations be performed for each valve application to determine the temperature drop of the fluid medium through the valve orifice, and;

2) The orifice be sized to prevent medium freezing.

One unique solution to the problem of medium freezing is to heat the downstream side of a valve with alpha particles. Laboratory tests have demonstrated the feasibility of this approach. The alpha radioisotope had an effective life of 10 years.

Not all valve applications require rapid or precise response rates. Typical response requirements are 5 to 50 msec for liquid bipropellant shutoff valves* and 12 to 200 msec for gaseous shutoff valves. Abnormal functional forces or inadequate electromagnetic or spring forces can adversely affect response rate. These problems have been discussed previously. A requirement for shorter response time than is actually needed will decrease operating life.

Monitoring solenoid valve position is desirable on both manned and unmanned missions for confirmation of system operation and status. Position indication for solenoid valves can be obtained from the transient current and voltage characteristics of the actuator coil. However, in some cases the current and voltage time traces are not properly interpreted resulting in rejection of the valve during qualification tests. Other position indication techniques utilize a mechanical linkage coupling a position transducer device to the linkage. On or off status may also be confirmed by precisely located switching devices coupled to the linkage. Many actuators do not couple to the valve mechanism through a mechanical linkage and conventional position indicating and measuring devices cannot be used.

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*Bipropellant valve - Any valve possessing two valving units in separate, isolated channels and actuated by a single actuator.

Lewis Research Center (Ref. 4) disclosed a circuit that detects the change in impedance of a solenoid coil caused by movement of the valve stem plunger. The impedance change unbalances a high frequency bridge, providing a signal for voltage-level detection circuitry that lights a lamp. A capacitor and resistor serves to null the bridge circuit for a given plunger position.

Another detection circuit can be hooked up to a second coil of a double-latching solenoid valve to determine whether the valve is open, closed, or at some intermediate position. If the detection circuits are identical, neither lamp will light if the valve is in an intermediate position. No valve modifications are necessary since the detectors connect directly to the coils; however, transient suppression is required to prevent damaging voltage spikes.

Parker Hannifin determines latching solenoid valve position by two methods: (1) a position switch which obtains its position from the armature, and (2) a magnetic reed switch which actuates when the upper solenoid air gap goes to zero. The magneto-motive force then becomes large enough to actuate the contact blades and provide electrical continuity.

d. Pressure Regulator Operates Out of Limits - Regulator drift out of limits can be caused by relaxation or creep of springs, holes in the actuation diaphragm or bellows, or contamination causing sticking of regulator parts or scoring the valve seat.

A rapid response rate is generally desired to keep the downstream pressure within design limits despite any rapid changes in flow rate. A low reference spring force is generally desired to allow a rapid response rate; however, high reference spring forces are required with the large sensing areas required for rapid response rate. A careful tradeoff is required to obtain the desired response rate within weight limitations.

Because a pressure regulator becomes heavier as the working and peak pressures increase, a point is reached where a pilot pressure regulator is lighter and controls better. The more complex piloted pressure regulator is inherently less reliable than the non-piloted types. It is therefore recommended that non-piloted pressure regulators be employed where design parameters permit.

Excessive overshoot can damage the system. The vent size on piloted pressure regulators determines the overshoot. The smaller the hole, the greater the overshoot; however, the larger the hole, the greater the loss rate of control fluid. Hence:

1) Allow the maximum overshoot consistent with system design to conserve the control fluid;

2) Provide sufficient control fluid to complete the mission, and;

3) Develop and use a sealed pilot system (preferable).

2. Design

a. Selection Criteria - The selection of a pressure regulators valve assembly for a particular application results from a detailed analysis of the specific requirements as well as the basic design characteristics of potential component types. The analysis is a systematic form of compromise wherein advantages and disadvantages of the different types are weighed against the specific requirements to permit an optional selection. This common approach to the pressure regulator and valve selection is basic to almost every design situation. Thus, it is apparent that the process is only as effective as the basic design criteria utilized to provide for consideration of all significant design factors. Any oversight in a basic design criterion could easily result in selecting a non-optimum component type with its attendant problems. Table 4 shows an idealized design factor list for solenoid valve and pressure regulator long-life assurance. Table 5 presents a design selection chart and a rating matrix of design parameters versus functional and environmental parameters from Ref. 5. Some significant design criteria are presented below:

1) *Flow Media* - The flow media is a basic design criterion and is a major factor in selecting the pressure regulator or valve to be used. Knowledge of the flow media (liquid, gas, or two-phase fluid) is essential because valve types vary in sealing capability with the fluid state. For example, a poppet valve generally will seal a light gas (i.e., helium) better than a butterfly valve. Additionally, the velocity of the media also must be examined because of flow of high velocity gases tends to erode seals. In selecting a valving assembly for such an application, consideration must be given to how well the main seat or sealing surfaces are protected from this erosion. This is particularly true for hot gases or highly-reactive gases. Material compatibility is of basic concern because material restrictions resulting from use of a particular gas could preclude consideration of a particular valve assembly. Flow media also can limit the valve assembly that can be considered as a result of contamination or the abrasive qualities of the medium. For example, a valve design

Table 4 Design Factors for Long-Life Assurance Part/Component: Solenoid Valve and Pressure Regulator

DESIGN FACTOR	REMARKS
Dynamic Seals	To meet required long-term leakage requirements, no dynamic seals other than bellows should be used (i.e., O-rings and omniseals are unacceptable).
Static Seals	Welded joints in place of captive types of seals are required for external leak paths and are also desirable for internal leak paths.
Main Poppet Force Margins (Open and Closed)	Very conservative force margins must be designed into the valve to insure operation at extremes of temperature, operating voltage, and dynamic loading, and in the presence of contamination and corrosion.
Microswitch Actuator Force Margins	Again, conservative force margins eliminate the possibility of contact chatter or, in the worst case, an erroneous signal.
Clearance between Sliding Parts	Contamination of the size that will pass through upstream filters must not cause jamming (i.e., clearances must be greater than the maximum dimension of such particulates). Avoid sliding parts, if possible.
Seat Configuration and Material	A soft seat (e.g., Teflon) is more tolerant of contamination than a lapped metal seat but must be carefully designed to minimize degradation from impact and scrubbing loads and to eliminate cold flow.
Coil and Electronic Part Potting	The potting material and configuration should be designed so thermal expansion and contraction will not damage or break connections.
Electrical Overloading	The design should be such that the inadvertent application of voltage for a period of time in excess of nominal design limits does not significantly degrade reliability. To prevent excessive coil overheat use latching solenoids to hold the valve open (or close) for long periods.
Faying Surfaces	Faying surfaces that could trap particulates during fabrication and release them during operation should be eliminated.

with inherently low sealing forces and significant rubbing action would not be a desirable design selection for a slurry medium. Flow media also must be considered in terms of freezing point and whether entrapped fluid could subsequently freeze, resulting in high expansion pressures or damage caused by icing effects.

2) *Pressure* - Pressure is a basic design criterion in selecting a valve type. Operating pressure not only establishes the basic structural requirements for the design, but eliminates some valve assemblies from consideration. For example, use of a gate-valve in a high-pressure system is not feasible because of its unbalanced pressure characteristics and the large actuation forces resulting from higher pressures. Similarly, the use of a ball valve or butterfly valve in a large line size at high pressure will result in prohibitive bearing loads, seal loads, and actuation forces. The operating pressure also must be examined from the transient aspect, as well as the static condition. If a valve assembly design is susceptible to pressure damage in the intermediate open positions, a high opening or closing transient pressure could cause seal failure.

In the case of the MOOG valve on the MMC Transtage Hydrazine ACS, a change of design was necessary because the engine produced fairly large tailoff spikes that hit the valve seat in the closed position. These spikes were of a magnitude of at least 1000 psi, and had a rise rate of over 10^6 psi/sec. The original seat design was of unsupported (enclosed) Teflon that was exposed to tensile load by the engine spikes. Therefore, these spikes knocked large pieces of the seat out and resulted in very serious leaks.

Two solutions were incorporated, first the seat was redesigned so that the Teflon became fully encased and would be loaded in compression by engine spikes. Secondly, the rocket engine injector was redesigned resulting in decreased pressure spikes. Should the operating pressure be compatible with the valve in terms of structural strength for static, as well as transient conditions, the number of times the pressure will be applied also must be considered. Repeated cycling of high pressure upon a valve assembly can result in fatigue failures.

3) *Flow Rate and Pressure Drop* - Flow rate and pressure drop must always be considered in the valve and pressure regulator selection. The required flow rate and the maximum allowable pressure drop will establish the required size for each of the designs. Once the size

APPLICABILITY			PERFORMANCE WITH PROPELLANTS																									
			GASES								EARTH STORABLES							SPACE STORABLES											C
			HYDROGEN	HELIUM	NITROGEN	OXYGEN	ARGON	CO_2	PROPELLANT BOILOFF	COMBUSTION PRODUCTS TO 1500°F	MON 10	NITROGEN TETROXIDE	AEROZINE 50	HYDRAZINE	MONOMETHYLHYDRAZINE	PENTABORANE	UDMH	CHLORINE PENTAFLUORIDE	CHLORINE TRIFLUORIDE	NITROGEN TRIFLUORIDE	NITRYL FLUORIDE	OXYGEN DIFLUORIDE	PERCHLORYL FLUORIDE	TETRAFLUOROHYDRAZINE	AMMONIA	DIBORANE	HYBALINE A[5]	LPG[1]	FLOX (80% LF_2, 20% LO_2)
FM	VALVE TYPE	FLOW METERING	3	3	3	3	3	3	3	2-3	3	3	3	3	3	3	3	3	3	3	3	3[d]	3[d]	3	3	3	3	3	3[d]
S	VALVE TYPE	SHUTOFF	2[e]	2[e]	3	3	3	3	2-3	2-3	3	3	3	3	3	2[e]	3	1-2[e]	1-2[e]	1-2[e]	1-2[e]	1[e]	2[e]	1-2[e]	3	2[e]	3	2[e]	2[e]
V	VALVE TYPE	VENT OR RELIEF	2[e]	2[e]	3	3	3	3	2-3	2-3	3	3	3	3	3	3	3	2	2	2	2	2	2	2	3	3	3	2[e]	2[e]
CR	VALVE TYPE	COLD GAS REGULATOR	2[e]	2[e]	3	3	3	3	2-3	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	3	NA	NA	2-3	2-3
HR	VALVE TYPE	HOT GAS REGULATOR	2-3	2-3	2-3	U	2-3	2-3	NA	2	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	3	NA	NA	3	U
LF	VALVE TYPE	LIQUID FILL OR DISCONNECT	2	2	2-3	2-3	2-3	NA	2-3	NA	2[b]	2[b]	2[b]	2[b]	2[b]	2[b]	2[b]	2[b]	2[b]	2[b]	2[b]	2[b]	2[b]	2[b]	2[b]	2[b]	3	1[c] 2[b]	1[c]-2[b]
PF	VALVE TYPE	PNEUMATIC FILL OR DISCONNECT	2-3[b]	2-3[b]	2-3[b]	2-3[b]	2-3[b]	2-3[b]	2-3[b]	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	2[b]	NA	NA	2-3[b]	2-3[b]
S	CLOSURE	BALL (POLYMERIC SEAL)	3	3	3	3	3	3	3	1	2	2	3	3	3	3	3	1-2	1-2	U	U	1-2	1	U	3	U	3	3	2
S,V,CR, HR, LF, PF	CLOSURE	POPPET	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	2-3	2-3	2-3	2-3	2	3	2-3	3	3	3	3	3
FM, S	CLOSURE	BUTTERFLY	3	3	3	3	3	3	3	3	2	2	3	2	2	3	2	1-2	1-2	1-2	1-2	1-2	1-2	1-2	3	3	3	3	1-3
S	CLOSURE	BURST DIAPHRAGM	3	3	3	3	3	3	3	2	3	3	3	3	3	3	3	U	U	U	U	U	U	U	3	3	3	3	U
S,V,CR,HR, LF, PF	POLYMERIC SEALS	VALVE CLOSURE	2-3	2[e]	3	3	3	3	3	1-3	2	2	2	2	2	2	2	1	1	U	U	1	1	U	3	U	U	1-3[e]	1-2[e]
ALL	POLYMERIC SEALS	STATIC SEALS	2-3	2-3	3	3	3	3	3	1-3	2	2	3	3	3	2	3	1-2	1-2	1-2	1-2	1-2	1-2	1-2	3	U	3	2-3[e]	2-3[e]
ALL	POLYMERIC SEALS	DYNAMIC SEALS	2-3	2[e]	3	3	3	3	3	1-3	2	2	3	3	2	3	3	1	1	U	U	1	1	U	3	U	U	1-3[e]	1-3[e]
S,V,CR,HR,LF,PF	METAL SEALS	VALVE CLOSURE	1-2[e]	1-2[e]	1-2[e]	1-2[e]	1-2[e]	1-2[e]	1-2[e]	1-2[e]	3	3	3	3	3	3	3	2-3	2-3	2-3	2-3	U	U	2-3	3	3	3	1-2[e]	2[e]
ALL	METAL SEALS	STATIC SEALS	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	2-3	2-3	2-3	2-3	2	2	2-3	3	3	3	3	3
ALL	METAL SEALS	DYNAMIC SEALS	1-2[e]	1-2[e]	3	3	3	3	3	3	3	3	3	3	3	3	3	2-3	2-3	2	2	2	2	2	3	3	3	3	2[e]
S	ACTUATORS	SOLENOID	3	3	3	3	3	3	3	1	2-3	2-3	3	3	3	3	3	3	3	U	U	U	U	U	3	3	3	3	U
FM, S	ACTUATORS	PNEUMATIC	3	3	3	3	3	3	3	3	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	3	NA	NA	3	1-3
FM, S	ACTUATORS	HYDRAULIC	NA	NA	NA	NA	NA	NA	NA	NA	2	2	2	2	2	3	2	3	3	U	U	U	U	U	3	3	U	3	1-3
FM, S	ACTUATORS	ELECT. MOTOR	3	1-3	3	3	3	3	3	1	2	2	2	U	U	U	U	U	U	U	U	U	U	U	3	U	U	U	U
S	ACTUATORS	SQUIB	3	3	3	3	3	3	3	1	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA
ALL	DRIVE MECHANISMS	SCREW DRIVES	3	3	3	3	3	3	1-3	1-3	U	U	1[f]	1[f]	1[f]	1[f]	1[f]	U	U	U	U	U	U	U	1[f]	1[f]	U	2	U
ALL	DRIVE MECHANISMS	BALL SCREW DRIVES	3	3	3	3	3	3	2-3	2-3	U	3[f]	1[f]	1[f]	1[f]	1[f]	1[f]	U	U	U	U	U	U	U	1[f]	1[f]	U	2	U
ALL	DRIVE MECHANISMS	LINEAR BALL DRIVES	3	3	3	3	3	3	2-3	2-3	U	3[f]	1[f]	1[f]	1[f]	1[f]	1[f]	U	U	U	U	U	U	U	1[f]	1[f]	U	2	U
ALL	DRIVE MECHANISMS	GEARED	3	3	3	3	3	3	2-3	1-2	U	3[f]	1[f]	1[f]	1[f]	1[f]	1[f]	U	U	U	U	U	U	U	1[f]	1[f]	U	2	U
ALL	DRIVE MECHANISMS	PISTON	3	3	3	3	3	3	3	1-3	3	3[f]	1[f]	1[f]	1[f]	1[f]	1[f]	U	U	U	U	U	U	U	1[f]	1[f]	U	3	U
ALL	BEARINGS AND BUSHINGS	BALL-ROTARY	3	3	3	3	3	3	3	2	U	3[f]	1[f]	1[f]	1[f]	1[f]	1[f]	2-3	2-3	U	U	U	U	U	1[f]	1[f]	2[f]	3	U
ALL	BEARINGS AND BUSHINGS	POLYMERIC	3	3	3	3	3	3	3	1	1	1	U	U	1	U	U	1	1	U	U	1	1	U	3	U	U	3	1-2
ALL	BEARINGS AND BUSHINGS	METAL	2[f]	2[f]	2[f]	2[f]	2[f]	2[f]	3[f]	1	U	3[f]	1[f]	1[f]	1[f]	1[f]	1[f]	2-3	2-3	U	U	U	U	U	1[f]	1[f]	2[f]	U	1-2
ALL	BEARINGS AND BUSHINGS	FLEXURE PIVOTS	3	3	3	3	3	3	3	2-3	3	3	3	3	3	3	3	U	U	U	U	U	U	U	3	3	3	3	U
ALL	HER-METIC SEALS	BELLOWS	3	3	3	3	3	3	3	2-3	3	3	3	3	3	3	3	2	2	2	2	3	3	2	3	3	3	3	2
ALL	HER-METIC SEALS	DIAPHRAGMS (METAL)	3	3	3	3	3	3	3	2-3	3	3	3	3	3	3	3	2	2	2	2	3	3	2	3	3	3	3	2
ALL	LUBES	DRY FILM	3	3	3	3	3	3	3	1-2	U	2	2-3	2	2	2	2	1	1	1	1	1	1	1	2-3	U	U	3	1
ALL	LUBES	LIQUID	3	3	3	3	3	3	1-2	1	1	1	2	2	1	1-2	2	1	1	1	1	1	1	1	1	U	U	1	1
ALL	SPRINGS	COIL	3	3	3	3	3	3	3	2-3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3
ALL	SPRINGS	BELLVILLE	3	3	3	3	3	3	3	2-3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3
ALL	SPRINGS	LIQUID	3	3	3	3	3	3	3	1	U	3[h]	3[h]	2[h]	3[h]	3[h]	3[h]	2[h]	2[h]	U	U	U	U	U	U	3[h]	U	1	U
FM, S, CR, HR	FITTINGS AND FASTENERS	AN FLARE AND MS FLARELESS	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	1[i]	1[i]	U	U	2	2	U	3	3	3	3	2
ALL	FITTINGS AND FASTENERS	WELDED	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	2	2	2-3	2-3	3	3	2-3	3	3	3	3	3
ALL	FITTINGS AND FASTENERS	BOLTS AND SCREWS	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3
ALL		FILTERS	3	3	3	3	3	3	3	1-3	3	3	3	3	3	3	3	2	2	1-2	2	2	2	2	3	3	3	3	2

Table 5 Valve Component Rating Analysis Chart (From Ref. 2)

HARD CRYOGENICS			METALLIZED GELS			NON-METALLIZED GELS			FUNCTIONAL PARAMETERS														ENVIRONMENTAL PARAMETERS	SPACE RADIATION						
LIQUID FLOURINE	LIQUID OXYGEN	LIQUID HYDROGEN	N_2H_4	UDMH	MMH	N_2O_4	ClF_3	MAF-MIXED AMINE FUELS	TEMPERATURE HIGH TO 2000°F	TEMPERATURE LOW	STERILIZATION 300°F 60 HRS. 6 CYCLES	POWER REQUIREMENTS	OPERATING LIFE 1000 CYCLES	CONTAMINATION	RESPONSE	VIBRATION AND SHOCK	LEAKAGE	SPACE MAINTAINANCE					ON BOARD NUCLEAR POWER SOURCES	VAN ALLEN BELT	SOLAR FLARE	INDUCED BREMSTRAHLUNG	SPACE VACUUM	ZERO G	METEROIDS	PROD. OF COMBUSTION (TURN AROUND)
3^{d}	3^{d}	3^{d}	1	1	1	2	U	U	U	3	2-1	2-3	3	3	3	3	2-3	U					2-3	3	3	2	3	3	2/1M	3
2^{c}	2^{c}	2^{e}	1	1	1	2^{c}	2^{c}	2^{c}	1	2-3	2-1	2	3	1-2	2	3	1-2	U					2	2	2	2	2-3	3	2/1M	2
2^{e}	2^{e}	2^{e}	U	U	U	U	U	U	2	2-3	2-1	3	3	1-3	2	2	1-2	U					2	2	2	2	2-3	3	2/1M	2
2-3	2-3	2-3	NA	NA	NA	NA	NA	NA	U	2-3	1	3	3	1-2	2	2	1-2	U					2-3	3	3	2	3	3	2/1M	3
U	U	2-3	NA	NA	NA	NA	NA	NA	2	U	2-1	3	3	1-2	2	2	1-2	U					2-3	3	3	2	3	3	2/1M	3
2^{b}	1^{a} 2^{b}	1^{a} 2^{b}	1	1	1	1	1	1	NA	2	2	NA	3	2	NA	3	1-2	U					2	2	2	2	2	3	2/1M	1-2
3^{b}	$2\text{-}3^{b}$	$2\text{-}3^{b}$	NA	NA	NA	NA	NA	NA	NA	3	2	NA	3	2	NA	3	1-2	U					2	2	2	2	2	3	2/1M	1-2
1	2	2	3	3	3	3	3	3	1	2-3	2-1	2	3	2-3	1-2	3	2-3	U					2	3	3	3	2	3	2/1M	3
3	3	3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	3	3	3	2	3	2	3	U					3	3	3	3	3	3	2/1M	3
-3	3	3	1-2	1-2	1-2	1-2	1-2	1-2	2-3	2-3	1	2	3	2-3	2	3	1-2	U					3	3	3	3	3	3	2/1M	3
U	3	3	3	3	3	3	3	3	2	3	1	3	NA	$1\text{-}2^{g}$	3	3	3	U					3	3	3	3	3	3	2/1M	3
$1\text{-}2^{e}$	$1\text{-}3^{e}$	$1\text{-}3^{e}$	1-2	1	1	2	1	2	1	2	1	NA	3	2-3	NA	3	3	U					1-2	1-2	1-2	1-2	2	3	2/1M	2
$2\text{-}3^{e}$	$2\text{-}3^{e}$	$2\text{-}3^{e}$	3	3	2	2	3	3	1	2-3	2	NA	2	3	NA	3	3	U					1-2	2	1-2	2	2	3	2/1M	3
$1\text{-}2^{e}$	2	2	1	1	1	2	2	2	1	2-3	2	1-2	3	2-3	2	3	3	U					1-2	1-2	2	1-2	2	3	2/1M	2
$1\text{-}2^{e}$	$1\text{-}2^{e}$	$1\text{-}2^{e}$	2	2	2	2^{c}	2^{c}	2^{c}	2	2-3	3	NA	3	1	NA	1-3	1-2	U					3	3	3	3	2	3	2/1M	1-2
3	3	3	3	3	3	3	3	3	2-3	3	3	NA	1	3	NA	3	3	U					3	3	3	3	3	3	2/1M	3
$1\text{-}2^{e}$	$1\text{-}2^{e}$	$1\text{-}2^{e}$	U	U	U	3	3	3	1-2	2-3	3	2	2	1	2	3	1-2	U					3	3	3	3	1	3	U	1-2
1	2-3	2-3	U	U	U	U	U	U	1	3	3	2	3	2-3	2-3	2-3	NA	U					1-2	2	2	3	1-2	3	2/1M	3
2-3	2-3	2-3	NA	NA	NA	NA	NA	NA	3	3	3	2-3	3	2-3	2-3	2-3	1-3	U					3	3	3	3	3	3	2/1M	3
2-3	2-3	2-3	U	U	U	U	U	U	1	1-3	1	2-3	3	2-3	2-3	3	3	U					3	3	3	3	3	3	2/1M	3
U	U	U	U	U	U	U	U	U	1	2	1	2	3	1-3	2	3	NA	U					1-2	2	2	2	1-2	3	2/1M	3
NA	NA	NA	NA	NA	NA	NA	NA	NA	1	3	U	3	1	3^{g}	3	3	1	1					U	U	U	U	U	3	U	3
U	2	2	1	1	1	3^{f}	U	1^{f}	1	U	3	2	3	2	2	3	NA	U					2-3	2-3	2-3	2-3	2	3	2/1M	3
U	2	2	1	1	1	3^{f}	U	1^{f}	1-2	2-3	3	3	3	2	2-3	3	NA	U					3	3	3	3	2	3	2/1M	2
U	2	2	1	1	1	3^{f}	U	1^{f}	1-2	2	3	3	3	2	2-3	3	NA	U					3	3	3	3	2	3	2/1M	2
U	2	2	U	U	U	U	U	U	U	2	3	2	3	2	2	3	NA	U					3	3	3	3	2	3	2/1M	2
3	3	3	U	U	U	1^{f}	1^{f}	1^{f}	3	3	3	2-3	3	2	3	3	2	U					3	3	3	3	2-3	3	2/1M	2-3
U	2-3	2-3	1	1	1	3^{f}	U	1^{f}	2	2-3	3	3	3	2	NA	3	NA	U					3	3	3	3	2	3	2/1M	2
1	2-3	2-3	1	1	1	U	U	U	1	2-3	2	2	2-3	3	NA	3	NA	U					3	1-2	2	2	2	3	2/1M	2-3
U	2	2	1	1	1	1^{f}	1^{f}	1^{f}	1-2	2-3	2-1	2	2-3	1-2	NA	3	NA	U					3	3	3	3	1-2	3	2/1M	1-2
U	3	3	3	3	3	3	3	3	U	3	3	3	3	3	NA	3	NA	U					3	3	3	3	3	3	2/1M	3
U	3	3	3	3	3	3	3	3	1-2	3	3	2	3	2	2	2	3	U					3	3	3	3	3	3	2/1M	3
U	3	3	3	3	3	3	3	3	2	3	3	2	3	2	3	3	3	U					U	3	3	3	3	3	2/1M	3
1	2	2	1-2	1-2	1-2	1-2	1	U	1-2	2	3	2	3	2	NA	2	NA	U					U	U	U	U	2	3	2/1M	2-3
1	1	1	1	1	1	1	1	1	1	1	2-1	2-3	3	2-3	NA	2-3	NA	U					2	2	2	2	2	1-3	2/1M	2-3
3	3	3	3	3	3	3	3	3	1-2	3		NA	3	3	2-3	2	NA	U					3	3	3	3	3	3	2/1M	3
3	3	3	3	3	3	3	3	3	1-2	3	1	NA	3	3	2-3	3	NA	U					3	3	3	3	3	3	2/1M	3
U	U	U	U	U	U	U	U	U	1	1	3	NA	3	3	2-3	3	1-2	U					U	U	U	U	2	3	2/1M	3
1^{i}	3	3	3	3	3	3	3	3	1	3	1	NA	1-2	2	NA	2	2-3	2					3	3	3	3	3	3	2/1M	3
2	3	3	3	3	3	3	3	3	3	3	1	NA	1	1-3	NA	3	3	U					3	3	3	3	3	3	2/1M	3
3	3	3	3	3	3	3	3	3	3	3	1	NA	1-2	3	NA	2	NA	U					3	3	3	3	3	3	2/1M	3
1-2	3	3	NA	NA	NA	U	U	U	U	3	1	NA	1-3	1	NA	2	NA	U					3	3	3	3	3	3	2/1M	3

LEGEND:

RATING CHARACTERISTICS
- 1- POOR
- 2- FAIR
- 3- GOOD
- U- UNAVAILABLE INFORMATION
- NA- NOT APPLICABLE

UNMANNED (DIGIT ONLY)

2 1M MANNED

VALVE DESIGNATION
- FM- FLOW METERING
- S- SHUTOFF
- V- VENT OR RELIEF
- CR- COLD GAS REGULATOR
- HR- HOT GAS REGULATOR
- LF- LIQUID FILL OR DISCONNECT
- PF- PNEUMATIC FILL OR DISCONNECT

NOTES:
- a) FREEZING OF CONDENSED MOISTURE AT INTERFACE OF DISCONNECT
- b) RATING FOR SERVICE IN VACUUM ENVIRONMENT
- c) RATING FOR SHUTOFF VALVE EXPOSED TO SPACE VACUUM
- d) NON CAVITATING FLOW CONTROL
- e) RATING BASED ON LEAKAGE CONTROL
- f) RATING BASED ON PROPELLANT LUBRICATION DATA AT LOW LOADS, SHORT DURATION
- g) CONTAMINATION GENERATOR
- h) PROPELLANT USED AS COMPRESSIBLE FLUID
- i) RATING BASED ON POOR RELIABILITY ASSOCIATED WITH PASSIVATION
- j) LIQUIFIED PETROLEUM GAS, METHANE, PROPANE, BUTENE
- k) α, β, γ, NEUTRONS RADIOISOTOPE NUCLEAR POWER SOURCES
- m) RATING BASED ON FORMATION OF CLOGGING MATERIAL

requirement for each type of valve is established, several valve types generally can be eliminated from consideration. For example, the use of a butterfly valve in a 1-in. line size or smaller system is impractical because of its high pressure drop characteristics which results when it is reduced to this size. Conversely, the use of a ball valve in large line sizes at high pressure also is impractical because of the resultant valve weight. Valve type selection also is affected by whether or not the flow rate is a steady-state requirement (i.e., in a shutoff valve) or a variable requirement (i.e., in a throttling valve). Generally, this is true of ball valves, gate valves, and blade valves. The transient flow characteristics of a shutoff valve during the opening and closing cycles could impose further limitations upon its use in a specific application.

4) *Leakage* - Leakage must be considered in terms of both internal leakage and external leakage. The more stringent the leakage requirements, the more difficult and costly it will be to achieve success. For example, an unrealistic, low leakage limit could eliminate consideration of the optimum type of valve. Leakage limits should be established as the most realistic values that will prevent depletion of the flow media, damage to equipment, danger to personnel, or failure of a mission objective. Once it is assured that reasonable leakage limits have been established, the various types of valve assemblies can be evaluated based upon their inherent sealing capabilities. An important consideration in this evaluation is the great variance in leakage rates resulting from design details and cycle life. The relative sealing capability of the various types of valving elements require thorough examination in relationship to the cycle life of the intended application.

Most of this discussion on internal leakage also applies to external leakage. Included in functional problems is the possibility of leakage past the seals of the electromagnetic plunger into the solenoid area. The problems of leakage past the plunger seals is not as severe as those for the poppet and seat because impact loads and seal design including redundancy can be employed more easily with plunger seals than can be employed with valve seats and poppets. It is suggested that plunger seals be avoided.

5) *Life* - Each valve or pressure regulator requires evaluation in terms of the cycle life and storage life needed for the intended application. The cycle life is defined as the maximum number of operations (open and closed or through the throttling range) required that can be accomplished without exceeding any of the basic sealing or functional requirements for the valve assembly. To adequately evalute the valve assembly in terms of cycle life capability requires a thorough knowledge of the state-of-the-art because cycle life for a particular type varies radically as size or test parameters (i.e., temperature, pressure, actuation rate, and fluid media) are altered. Also, the cycle life varies greatly within a valve assembly as a result of design details (i.e., seal loads, seal materials, seal shapes, and seal retention). In addition to the criteria related to the total number of required cycles and the total storage life, the cycling is required during the life of a valve, as well as whether or not the valve is dry or exposed to the flow media during any idle periods. Proper design also calls for an evaluation of what affect the duty cycle has upon the cycle life for each of the valve assembly under consideration.

6) *Other Design Criteria* - The preceding design criteria are the more obvious factors to be considered when selecting a valve assembly design. A number of other factors also require consideration. The environment requirements are typical of these. The operating temperature range restricts the use of various seal materials and results in a further limitation. For example, sealing of hot gases cannot be accomplished using plastics or elastomers. Temperature also can influence design selection from a transient aspect because the use of some designs are not desirable where thermal shock is involved or where sealing is required during a change in temperature. The environment also must be considered with respect to vibration, acceleration, shock levels. This is particularly important in selecting the valve and actuator combination.

Exposure to vacuum constitutes additional design criteria. Vacuum exposure limits sealing techniques by restricting the use of materials. It also affects design selection by restricting the use of lubricants because many lubricants will boil-off at hard-vacuum levels. The necessity for removing the flow media to allow for safe handling or storage also must be examined. If a valve is to be utilized in toxic fluids, the basic design must provide for adequate fluid removal to allow safe handling during rework or use. A need for the complete fluid removal could exist as a result of using corrosive or highly-reactive fluids having limited times that the valve elements can be exposed to them.

Any detailed design of the valve assembly should include the method of actuation, method of operation, type of seat (hard or soft), seat materials, poppet geometry, poppet materials, type of fluid handled and many other variables.

Compatibility with the environment is a life-limiting factor. Most propulsion systems operate in two modes only; maximum or zero thrust. Hence, the pressure regulator used in this applicaton is normally almost fully open, or closed with minimal cycling. Hence, the pressure regulator problems may be time dependent rather than cycle sensitive. Extreme care must be exercised in the selection of valve seat, seal, and dielectric materials to ensure functioning after an exposure to space environments for 10 years.

An area of concern is the effect of elastomers in valves which are idle for long periods. O-rings left in contact with metal for long periods will "vulcanize" to the metal. The following guidelines are recommended:

1) Always restrain Teflon on three sides. Kel-F has even more tendency than Teflon to shear;

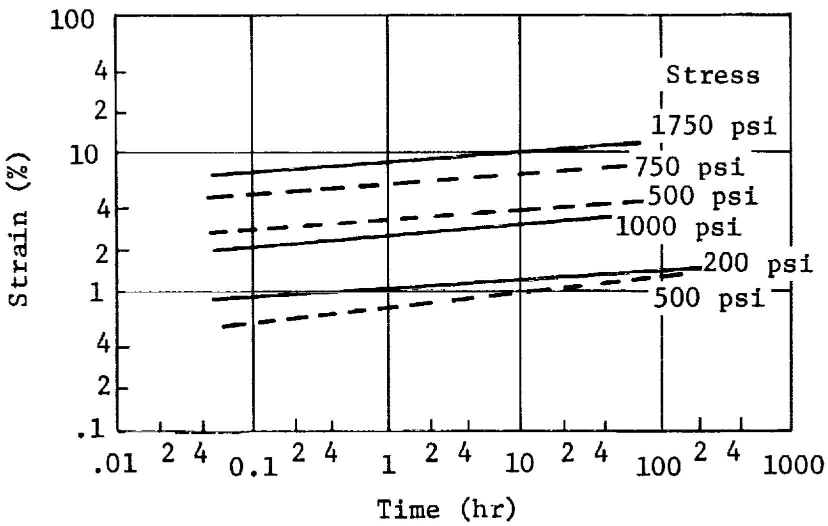
2) Keep Teflon stress below 10,000 psi to avoid extruding, and;

3) Use narrow seats to minimize the probability of contamination causing leakage.

Jack Potter (MSFC) indicated special attention is given Teflon in valve seat design to prevent cold flow. The suggested control methods to be used are: (1) heat stabilization, (2) contain material on three sides, (3) use a ball seat that is less sensitive to cold flow. He warned that Teflon should not be used in high pressure oxygen systems because of the flammability hazards.

The life of plastic seats can be adversely affected by creep. Creep is defined in Reference 6 as the total deformation under stress after a specified time in a given environment beyond that instantaneous strain which occurs immediately upon loading. Independent variables which affect creep are time, temperature, and stress level.

An initial strain or deformation occurs instantaneously as a load is applied to Teflon. Following this initial strain is a time period during which the part continues to deform at a decreasing rate. Creep data over a wide range of temperatures are plotted for compressive loading in Figure 9.



Dashed Line - Data Obtained at 212°F
Solid Line - Data Obtained at 73°F

Figure 9 Total Deformation versus Time under Compressive Load

The creep properties of plastics under compressive load and the deformation with time are discussed in Reference 7. Table 6 shows that TFE's tendency to creep can be suppressed several hundred percent by the addition of appropriate fillers.

Most creep occurs during the first 24 hours. After this, creep falls off and the deformation curve stabilizes. Extrapolation of Table 6 deformation measurements at 24 and 100 hours should yield a rough estimate of long-term creep. Of course, a temperature rise reduces the compounds' creep resistance.

Once the deformation-producing load is removed, both filled and unfilled TFE resins show a plastic recovery; that is, the test sample tends to return to its original shape. At room temperature, this recovery amounts to about 50% of the total deformation (if yield strain is not exceeded). Table 6 defines "permanent deformation" as total deformation less room temperature recovery after 24 hours.

The survey of manufacturers indicated they determine solenoid opening (or closing) force margins in terms of voltage. Normal spacecraft voltage is 28 ± 4 VDC and the solenoid pull in voltage ranges from 15 to 18 VDC. A force margin may then be computed using these data, a force margin goal of 300% is suggested. In addition to the valve opening force, the closing force should be such that the valve seats are not hammered out because of excessive forces or a short valve response time.

General long-life problems and alternative solutions have been presented; however, long life cannot be obtained by merely applying general principles. Each specific solenoid valve and pressure regulator application and its environment must be carefully analyzed to obtain long-life with a high probability of success. Only vendors with a history of success in space applications should be considered to obtain long life. A discussion of the factors in Table 7 is presented in Ref. 5 and 8.

b. *Results of Industry Survey* - A general survey of the manufacturers and users of solenoid valves and pressure regulators was conducted to ascertain the expected life of existing hardware and to identify features which would enhance valve life in new designs. The consensus answers to the questions posed are reported in Table 8. Table 9 presents the results of a second survey, the Specific Component Survey. Comments from both surveys are noted throughout the study.

Table 6 Compressive Creep Properties of Typical Filled TFE Compositions (From Ref. 5)

Property		Un-filled TFE	15% Glass Fiber	25% Glass Fiber	15% Graphite	60% Bronze	20% Glass 5% Graphite	15% Glass 5% MoS_2
% Deformation @ 78°F, 2,000 psi., 24 hr.	MD[2]	4.3	8.3	7.1	8.1	6.0	6.8	6.9
	CD[3]	6.7	13.4	7.5	9.5	5.3	6.7	7.1
% Permanent deformation[1]	MD	7.9	4.1	3.9	4.4	2.5	4.9	3.8
	CD	8.4	9.0	4.6	5.3	2.3	3.9	3.9
% Deformation @ 78°F, 2,000 psi., 100 hr.	MD	16.3	12.6	8.9	10.1	6.1	7.6	7.8
	CD	18.7	14.9	9.4	11.5	6.4	8.7	8.1
% Permanent deformation	MD	8.8	6.0	4.4	6.4	2.5	5.8	5.6
	CD	9.1	7.9	5.6	7.3	2.5	5.9	5.5
% Deformation @ 500°F, 600 psi., 24 hr.	MD	30.1	16.6	10.6	16.0	10.6	11.3	9.6
	CD	32.8	27.7	27.8	15.4	8.4	12.2	10.9
% Permanent deformation	MD	17.4	11.9	4.9	12.0	7.1	8.4	6.4
	CD	19.2	16.2	17.9	10.8	4.9	8.4	6.8

1. After 24 hours recovery

2. MD = Parallel to the molding direction

3. CD = (Cross direction) perpendicular to molding direction.

Table 7 Factors Influencing Valve Selection and Design

Application	Valve Type	Construction	Electrical	Performance	Reliability	Environmental	Interface
Attitude Control Maneuvering Soft Landing Orbit Injection Steady State Pulse Mode Size Range	Piloted Direct Acting Balanced Unbalanced Actuator Torque Motor Solenoid Pneumatic Hydraulic Electric Motor	All-Welded Assembly Lubrication Small Dribble Volume Use a Mechanical Linkage Facilitate Cleaning Sliding Parts Seat Design Electrical Cables or Connectors Inlet Filters Instrumentation Materials	Coil Resistance Current Voltage Power Insulation Resistance Voltage Suppression Thermal Compensation Heat Generation Magnetic Flux Path Materials Temperature EMI	Internal Leakage External Leakage Inlet Pressure Pressure Drop Pull-in and Dropout Voltage Size Weight Flowrate Response Opening Closing Current Backpressure Positioning Accuracy Flow Characteristic	Failure Mode Service Life Shelf Propellant Service Cycle Environmental	Vibration Sinusoidal Random Acceleration Vacuum Temperature Sterilization Fungus Humidity Radiation Shock Transportation	Heat Soakback Voltage Suspension Contamination Levels Pressure Surges

Table 8 Results of the General Manufacturing/Agency Survey - Solenoid Valves and Pressure Regulator

QUESTIONS	CONSENSUS ANSWER
1. Do you manufacture (use) aerospace solenoid valves and regulators? Usage?	Yes, however the missions are relatively short duration.
2. What is the expected life of subject part?	Solenoid valves and pressure regulators cycle life ranges from 30,000 to 2,000,000 cycles. Most storage lives are called out as 5 years, however, aging of non-metallic parts are the only life limiting factor.
3. What failures have occurred? What are the failure causes?	System contamination resulting in internal leakage is the most mentioned failure mode. Other failure modes are: valve sticks, loss of coil, fails to open. Regulator problems are: sensing element capsule leakage, diaphragm or bellows leaking, damaged seats.
4. What solutions do you suggest for the above failure modes that would enhance the operational life or increase the probability of success?	Control system contamination, solenoid valve closing forces, and lower stress that valves and regulators are exposed to. Place filters upstream of critical components such as pressure regulators.
5. To achieve long life, what design features should be incorporated into the subject part for space applications?	Use welded body construction and weld components into the systems. Consider inflight maintenance of valves and regulators by using plug-in or screw-in concepts. Subject valves to lower stresses. Avoid the use of teflon due to swelling or cold flow.
6. How did you determine part life?	Analysis, FMEA's, life cycle tests and age sensitive materials. Cycle examine for wear and determine critical parts.
7. Did you test for specific failure modes?	If there is an indication from FMEAs or analysis of a different type of failure mode, then tests such as foreign matter tests, or operating a valve without an O-ring seal may be run to check for leakage.

Table 8 Results of the General Manufacturing/Agency Survey - Solenoid Valves and Pressure Regulator (cont)

QUESTIONS	CONSENSUS ANSWER
8. Is accelerated testing and margin testing used?	Accelerated tests are mainly cyclic and the margin tests are performed with pressure, temperature and vibration.
9. What process controls are necessary for long life?	Clean parts and their assembly in clean areas, 100% inspection of parts, 50 to 100 run-in cycles, log of pressure limits for regulators. Thermal cycles are run on customer's request.

Table 9 Specific Component Survey

COMPONENT - SOLENOID VALVE

PART DESCRIPTION AND USING PROGRAM	USER'S OPINION OF WHY PART WAS SUCCESSFUL	MANUFACTURER'S OPINION OF WHY PART WAS SUCCESSFUL	RECOMMENDED GUIDELINES IMPROVEMENTS, PROCUREMENT SPEC REQUIREMENTS, ETC.	RELATIVE IMPACT ON YIELD & COST OF INCORPORATING RECOMMENDATIONS
Helium Solenoid Valve used in LM Pressurization System. A candidate for Shuttle application.	1. Agrees with manufacturers comments. 2. Earlier problems have been corrected. Improvements in the position indicator and casting impregnation have been highly successful. 3. The latest configuration has an enlarged poppet to seat configuration and is very susceptible to contamination. This configuration has a user part number.	1. Solenoids with permanent magnet armatures are used to inherently provide very high latching forces. There are no separate devices to apply or release the latch. 2. No sliding fits and no sliding seals. The valve cannot stick or jam. 3. Low internal leakage is obtained by use of a teflon valve seat. 4. The valve poppet is pressure balanced to both inlet and outlet pressures, reducing valve actuation forces to less than 1/50 of that required for an unbalanced valve. 5. All welded construction assures zero external leakage. 6. Valve position is monitored without compromise of reliability by means of a magnetically-actuated reed switch.	Impose existing user specification. Life qualification to 20,000 cycles required for Shuttle application. User is confident of success. The earlier configuration has demonstrated 10,000 cycles.	User estimates that latest configuration will have a decrease in rejection rate of 80-90% compared to earlier configuration. Some non-recurring cost for life test. Minor increase in recurring cost.

Table 9 (cont)

COMPONENT - SOLENOID VALVE (Cont)

PART DESCRIPTION AND USING PROGRAM	USERS OPINION OF WHY PART WAS SUCCESSFUL	MANUFACTURER'S OPINION OF WHY PART WAS SUCCESSFUL	RECOMMENDED GUIDELINES IMPROVEMENTS, PROCUREMENT SPEC REQUIREMENTS, ETC.	RELATIVE IMPACT ON YIELD & COST OF INCORPORATING RECOMMENDATIONS
Helium Solenoid Valve (cont)		7. EMF spikes are clipped by the internal voltage surge suppression circuit. 8. Materials used are completely compatible with all storable propellants.		

Table 9 (cont)

COMPONENT - SOLENOID VALVE

PART DESCRIPTION AND USING PROGRAM	USERS OPINION OF WHY PART WAS SUCCESSFUL.	MANUFACTURER'S OPINION OF WHY PART WAS SUCCESSFUL	RECOMMENDED GUIDELINES IMPROVEMENTS, PROCUREMENT SPEC REQUIREMENTS, ETC.	RELATIVE IMPACT ON YIELD & COST OF INCORPORATING RECOMMENDATIONS
Solenoid Valve used in the TIIIC Transtage Pressurization system.	1. A highly reliable valve when contamination levels are within specification. 2. Soft seat. 3. Qualified for 100K cycles. 4. Ten micron inlet filter.	1. Five dash number changes have been made to valve since 1965. (See Column 5). The vendor now feels that he has a good valve design if contamination is controlled within specification.	1. Place filter at outlet side of valve to prevent contamination from entering valve during back flushing operation. 2. Careful control of contamination.	It is suspected that this valve failed on the first Titan Transtage flight. As a result of this failure the following changes were made: . Solenoid was increased in size. . Poppet was redesigned. . Contamination level requirement was tightened. Since then there have been no flight failures. However, there have been a considerable number of rejections in the leak test facility. About 90% of these rejections have been caused by contamination. Most of the contamination levels were out of specification for this valve and was interjected by back flushing. The vendor feels that a filter on the outlet of this valve would have prevented most of our rejections.

Table 9 (cont)

COMPONENT - SOLENOID VALVE (Cont)

PART DESCRIPTION AND USING PROGRAM	USERS OPINION OF WHY PART WAS SUCCESSFUL	MANUFACTURER'S OPINION OF WHY PART WAS SUCCESSFUL	RECOMMENDED GUIDELINES IMPROVEMENTS, PROCUREMENT SPEC REQUIREMENTS, ETC.	RELATIVE IMPACT ON YIELD & COST OF INCORPORATING RECOMMENDATIONS
Solenoid Valve used in the TIIIC Transtage Pressurization System. (cont)				• Some increase in non-recurring cost for requalification. Slight increase in recurring cost for incorporating the filter.

Table 9 (cont)

COMPONENT - CHECK VALVE

PART DESCRIPTION AND USING PROGRAM	USER'S OPINION OF WHY PART WAS SUCCESSFUL	MANUFACTURER'S OPINION OF WHY PART WAS SUCCESSFUL	RECOMMENDED GUIDELINES IMPROVEMENTS, PROCUREMENT SPEC REQUIREMENTS, ETC.	RELATIVE IMPACT ON YIELD & COST OF INCORPORATING RECOMMENDATIONS
Helium Check Valve used in LM Pressurization System	User agrees with manufacturer's comments; however, user stated that this valve should not be used on Shuttle until the internal leakage problem is corrected. With only two valves per LM vehicle, 145 failures occurred. These were predominantly internal leakage (133), of which the predominant cause was contamination (125). The user has concluded that the problems are caused by seat to poppet misalignment and contamination susceptibility. The latter problem has been corrected, but the former has not.	1. Sealant impregnated valve castings. 2. Materials tested to be compatible with fluid impurities. The test considered dynamic and static applications. Also considered were temperature, pressure and phase variations of fluid and corrosion susceptibility. 3. Use rolled-on threads. 4. Quad redundant check valve (4 elements) to prevent back flow of propellants or helium into pressurization system. 5. All welded body construction which minimizes leakage and reduces weight. 6. No metal-to-metal moving fits. 7. All internal dynamic and static sealing is accomplished by teflon seats and gaskets, thus obviating any compatibility problems.	The features listed in Column 3 are considered good for check valves. However, further study is needed relative to seat life, spring configuration and contamination susceptibility before a valve is selected for Shuttle. Another prevalent problem that requires solution is low flow flutter, which is more of systems problem than a component problem.	Unknown

Table 9 (cont)

COMPONENT - PRESSURE REGULATOR

PART DESCRIPTION AND USING PROGRAM	USERS OPINION OF WHY PART WAS SUCCESSFUL	MANUFACTURER'S OPINION OF WHY PART WAS SUCCESSFUL	RECOMMENDED GUIDELINES IMPROVEMENTS, PROCUREMENT SPEC REQUIREMENTS, ETC.	RELATIVE IMPACT ON YIELD & COST OF INCORPORATING RECOMMENDATIONS
Helium Regulator for LM	1. Simple, directly spring-loaded design. Achieves high accuracy with a large area sensing element and a low spring rate achieved by Belleville spring techniques. 2. No sliding seals in the regulator. Example: The inlet valve is "pressure balanced" by a unique compliant plate technique which eliminates the critical sliding seals traditionally used. 3. All metal, all welded construction provides zero leakage and compatibility with all common fluids. 4. Multiple ply bellows to provide: Redundancy against leakage and damping to improve stability and life. 5. Exhaustive piece part testing (e.g., Belleville Spring). 6. Redundant "O" ring seals.	1. Agree with user's comments. 2. Use of rolled threads. 3. Minimum dead end passages. 4. Sealant impregnated valve casting. 5. Metal-to-metal seats. 6. Ultrasonically cleaned parts. 7. Single contamination control specifications. 8. Deburring of all parts.	• Implement user's and manufacturer's comments. • <u>Recommended Changes</u> 1. A more precisioned poppet to seat manufacturing process. 2. Filtration specification (upstream and downstream) should be agreed upon by the valve manufacturer and user, and be controlled to this specification. 3. Demonstrate 1 million cycle qualification life test. 4. Perform 100 run-in cycles acceptance test. 5. Install soft seats in secondary valves. 6. Conduct contamination susceptibility test during development tests. Manufacturer has reservations about meeting the 1 million cycle life requirement of the Space Shuttle and, hence, recommends a life demonstration.	• Forty-one regulators were rejected by the user. Approximately 75% of the failures were internal leakage caused as follows: - 40% contamination. - 35% poppet to seat misalignment. • Some additional nonrecurring cost for incorporating recommended changes. Recurring cost increase should be negligible.

Table 9 (cont)

COMPONENT - PRESSURE REGULATOR

PART DESCRIPTION AND USING PROGRAM	USERS OPINION OF WHY PART WAS SUCCESSFUL	MANUFACTURER'S OPINION OF WHY PART WAS SUCCESSFUL	RECOMMENDED GUIDELINES IMPROVEMENTS, PROCUREMENT SPEC REQUIREMENTS, ETC.	RELATIVE IMPACT ON YIELD & COST OF INCORPORATING RECOMMENDATIONS
Helium Pressure Regulator The specific design has not been built, but has been proposed for an unknown program and is very similar to the valve used on VELA, Pioneer, Ranger, Mariner, Bio-Stellite, and AF Project 206. Recommended by manufacturer as a candidate for Shuttle OMS helium system. Component has never experienced failure of springs, screws, diaphrams or valve bodies on the above programs.	1. Series redundant valves built into one package, with a system of two valve packages in parallel resulting in redundancy in both the fail open and fail closed modes. 2. The series redundant valve package has two valve seats. The upstream seat is a hard seat and the downstream seat is a soft seat (teflon). 3. A backup metallic diaphragm with a piston type seal to provide redundancy in the sensing function.	1. Agree with user's comments. 2. Use poppets made of tungsten carbide with a titanium binder identified as Kenna-Metal K601, which has been proven for the past 18 years. 3. Sealant impregnated valve casting. 4. Vacuum melt bar stock. 5. Use rolled threads. 6. Deburr all parts. 7. Use single contamination control specification. 8. Ultrasonically clean parts and store clean parts in sealed bags. Use nylon or polyethylene bags.	Implement user's and manufacturer's comments. <u>Recommended Changes</u> 1. Conduct 1 million cycle qualification test. 2. Perform 50 to 100 cycle run-in acceptance test. 3. Perform contamination susceptability test during development. 4. Perform corrosion susceptability test and fluid compatibility test.	. Manufacturer would not speculate on estimated rejection rate. . User stated that the failure rate for this valve has been low. . Some increase in nonrecurring cost. Impact on recurring cost unknown.

Table 9 (cont)

COMPONENT - PRESSURE REGULATOR

PART DESCRIPTION AND USING PROGRAM	USERS OPINION OF WHY PART WAS SUCCESSFUL	MANUFACTURER'S OPINION OF WHY PART WAS SUCCESSFUL	RECOMMENDED GUIDELINES IMPROVEMENTS, PROCUREMENT SPEC REQUIREMENTS, ETC.	RELATIVE IMPACT ON YIELD & COST OF INCORPORATING RECOMMENDATIONS
Helium Pressure Regulator for Apollo A candidate for Shuttle OMS and RCS pressurization systems.	1. Series redundant valve package. 2. Spring loaded, direct acting pilot valve with pilot and main flow paths being parallel. This provides smooth proportional flow control over the entire range. 3. Regulator setting with addition or deletion of shims which prevents external adjustment. 4. Internal leakage is controlled by the use of plastic (Kynar) poppets and metal seats. 5. All internal-to-external joints are welded or brazed.	1. Agrees with the user's comments. 2. Similar manufacturing processes and testing to other manufacturers surveyed.	Implement user's and manufacturer's comments. <u>Recommended Changes</u> 1. Change material of O-ring from silicone to EPR rubber to provide long-term compatibility with N_2H_4. 2. Provide seamless bellows used in Minuteman application instead of edge-welded bellows to reduce failure rate for external leakage. 3. Perform the following tests: . 1 million life cycles, including 50 slam-start cycles. . Overshoot during slam-start with various downstream "blanket" pressures. . Control stability under all pertinent combinations on inlet pressure, unit and gas temperatures and flow transients.	. The manufacturer would not speculate on estimated rejection rate. . 29 failures experienced on Apollo during testing. About 80% of the failures were leakage due to contamination. . Some increase in non-recurring cost. Impact on recurring cost unknown.

Table 9 (cont)

COMPONENT - PRESSURE REGULATOR (Cont)

PART DESCRIPTION AND USING PROGRAM	USERS OPINION OF WHY PART WAS SUCCESSFUL	MANUFACTURER'S OPINION OF WHY PART WAS SUCCESSFUL	RECOMMENDED GUIDELINES IMPROVEMENTS, PROCUREMENT SPEC REQUIREMENTS, ETC.	RELATIVE IMPACT ON YIELD & COST OF INCORPORATING RECOMMENDATIONS
Helium Pressure Regulator for Apollo (cont)			• Fail open flow (i.e., flow limiter evaluation). • Performance of package, first with primary unit blocked open, and then with secondary unit blocked open.	

Table 9 (concl)

COMPONENT - PRESSURE REGULATOR

PART DESCRIPTION AND USING PROGRAM	USER'S OPINION OF WHY PART WAS SUCCESSFUL	MANUFACTURER'S OPINION OF WHY PART WAS SUCCESSFUL	RECOMMENDED GUIDELINES IMPROVEMENTS, PROCUREMENT SPEC REQUIREMENTS, ETC.	RELATIVE IMPACT ON YIELD & COST OF INCORPORATING RECOMMENDATIONS
Helium pressure regulator. This fifth generation component has been used on the following programs: . Gemini . Lunar Orbiter . Transtage . Minuteman III . MOL . Mariner 71 . Viking 75 This component is a candidate for Shuttle.	1. No dynamic seals. 2. Hard seat. 3. Direct acting (vs pilot operated). 4. Extensive flight history with associated qualification testing; no flight failures; two qualification test failures. 5. Excellent stability. 6. Internal filtration. 7. Uses only freon lubrication on propulsion threads and uses flare savers. 8. Shim adjustment.	1. Agree with user. 2. Deburr all parts. 3. Use ultrasonic cleaning.	. Use off-the-shelf for Shuttle with pressure adjusted as required and life demonstration for one million cycles. Must duplicate the system redundancy configuration in the life demonstration test. . High confidence that this component will pass the one million life cycle demonstration.	. Only two qual test failures on this valve have been experienced on the Mariner and Viking programs. Both failures were due to contamination (particles) induced into the system because of inadequate procedure. . It is speculated that any life problems that develop on this valve will result from a system design that will allow out-of-specification contamination or too low a flow rate. . No recurring cost increases. . Some non-recurring cost increases to perform life test. . These cost impacts assume that resizing is not required and that qualification by similarity is acceptable.

c. Alternate Approaches - This subsection presents the alternate components and methods which may fulfill the technical objectives or approaches to alleviate life limiting failure modes of present solenoid valves or pressure regulators.

1) *Present Designs* - A solenoid valve and a pressure switch (called a bang-bang system) can perform the same function as a pressure regulator. A comparison of the generic failure rates of a bang-bang system versus a pressure regulator indicates that the pressure regulator is somewhat more reliable.* The main reason for selection of a bang-bang system on the Titan III transtage was accuracy (approximately ±1.0 psi) which can't be accomplished within a single stage modulating regulator. Thus, a pressure regulator developed for a specific set of design constraint and environmental parameters should be more reliable than a bang-bang system. In long-life system designs, consider:

1) Using pressure regulators in lieu of a bang-bang system, and;

2) Employing an accumulator between states of pressure regulators (if used) to reduce the coupling effect.

Manual actuator overrides are not normally required on spacecraft valving, although they have been used in aircraft and missile ground support applications. The manual override capability must be considered in the evaluations of manned spacecraft systems where it may affect overall system reliability. Manually controlled valves have been used as primary and secondary controls in the life support systems on the Mercury and Gemini capsules. The use of manually controlled functions in these applications simplified overall system design and improved reliability. Further study is required on specific missions and systems to maximize the advantages of man's adaptive and analytic nature with respect to mission success probability.

If the particular system design permits, it may be possible to circumvent the valve internal leakage problem by:

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*This reliability comparison is for accomplishing a pressure regulation function only. If both pressure regulation and shutoff functions are required, the bang-bang system may be more reliable because the solenoid valve could perform both functions.

1) Employing integral burst discs in the valve, and;

2) Installing zero leakage ordnance valve(s) in series with the solenoid valve.

2) *Future Designs* - Possible future designs are listed in the following paragraphs. Sources of design details may be extracted from the reference and bibliography lists.

TRW (Ref. 5 and 8) is studying a radial shutoff valve. Eliminating external shutoff valves for rocket engines by incorporating a leakfree flow shutoff function within the venturi injector of a rocket engine is desirable. The radial shutoff valve can perform the shutoff function by either expanding an internal poppet against a circular seat or contracting an outer radial poppet against an internal seat. In either case, the seating force can be derived by an overtravel movement of the venturi or injector pintle actuator with no wiping movement between the poppet and seat. The advantages of this arrangement are:

1) Elimination of separate valve actuators;

2) Reduction in engine weight;

3) Long life because the poppet is seated in direct compression, and;

4) Bubbletight shutoff due to the high closure pressure necessary to seat a material such as Teflon.

A cone labyrinth valve developed by Smirra Development Company of Los Angeles, California, employs a novel concept of seat-surface contact. The flow control element of the valve accomplishes both throttling and shutoff by the use of two concentric sets of flexible metal blades. Throttling is accomplished by forcing capillary flow through the labyrinth created by valve closure; the sealing action results from the engagement of the resilient metallic sliding surfaces. Because of all the metallic contact, the probability of cold welding in a vacuum may be greater than for other types of valve seats. Some of the advantages of this design are:

1) High corrosion resistance afforded by the all metallic design;

2) Extreme temperature capability;

3) Seats are insensitive to contamination because impurities are scraped away;

4) Seal life is extended by self-sealing feature, and;

5) Sealing is made redundant by the use of multiple seats.

Studies of "wet seals" are being conducted at TRW Systems. A liquid/metal interface is introduced between the mating metal parts to fill leakage paths and to permit adequate sealing at seat stresses considerably below the yield strength of the material. See Ref. 9 for more details on this approach and others.

Goddard Space Center used thermoelectric cooling to freeze flowing liquids. This unusual technique stopped flow in less than 10 msec.

The use of electromagnetic forces to actuate the poppets has received some attention. The poppet is attached to a hinged mechanism that has permanent magnets attached physically opposite each other. The valve is opened by applying current in the proper direction to the electromagnets. The valve is normally latched closed magnetically. To hold the valve closed under vibration environments, "reverse" current can be applied to the electromagnets.

d. Expected Life - Reference 10 presents 37 proven valve designs including key parameters (such as the expected life) and salient features of the seat and closure mechanisms. These mature valves were employed in aerospace applications and included adaptations of all types of valves. Not all of the valves listed were solenoid operated. The valves were divided into five classifications, and the mean cyclic life of each classification was calculated. The mean cyclic life of each classification of "standard" quality aerospace valves is shown below. Since margin testing was apparently not performed on these valves (or none reported), the actual mean cyclic life may be higher than shown.

Valve Classification	Mean Cyclic Life (Cycles)
Flat Poppet Seats	8,125
Conical Poppet Seats	18,750
Spherical Poppet Seats	38,714
"Space" Quality Long Life	1,500,000
Solenoid Only - Composed of Previous Four Classifications	38,571

Eight of the valves had lives in excess of 50,000 cycles. One Clary Dynamic's solenoid valve with a spherical poppet seat had a 2 million cycle life. Selected characteristics of the valves listed in Ref. 10 are given in Table 10.

The consensus from the survey of manufacturers and users indicated that a spring return solenoid valve and pressure reducing regulator service life of 10 years without maintenance is currently possible. The cycle life estimates ranged from 30,000 to 4,000,000 for solenoid valves and from 30,000 to 10,400,000 for pressure regulators. The component types were those used in ground applications and on spacecraft such as Gemini, Apollo, and Skylab in the environmental control systems. These components are used in O_2 and N_2 gas systems with pressures that range from 5 to 3000 psi. From these data it is estimated (excluding high and low estimates) a 10 year service life and 500,000 cycles are currently possible with state-of-the-art solenoid valves and pressure regulators. A 10 year service life and 1,000,000 cycles may be obtainable with advanced state-of-the-art components.

Ref. 11 indicates that the mission requirements are not this severe. For example, the Mariner IV valves actuate about 9000 times per year. A typical planetary lander may require 10,000 valve actuations.

The state-of-the-art capability now exists to fulfill mission requirements for solenoid valve life for Skylab and Viking. These mission applications are all for less than 1 year. Solenoid valves for military space systems operating for periods of up to 5 years are currently within the state-of-the-art.

Table 10 Selected Valve Parameters for Comparison (from Ref 10)

Manufacturer	Type	Rating	Maximum Leak Rate	Life (Cycles)
Plug Valves				
1. Rocketdyne	3/4 in. N Rotating Cylinder	600 psig -65 to 160°F	15 scim	1,000
2. Rocketdyne	1.75 in. lox Ball Valve	1800 psig -320 to 160°F	20 scim	1,000
Flat Poppet Seats				
1. Rocketdyne	1 in. Relief Valve	1000 psig -30 to 160°F	2 scim of Air	1,000
2. Bendix	Pneumatic Thrust Controller	225 psig 30 to 160°F	0.08 sccm of N_2 at 170 psi	1,000,000
3. Rocketdyne	3/4 in. Pressure Regulator	5000 psig -320 to 500°F	25 scim of He	10,000
4. Weston Hyd. Ltd.	Three Way Solenoid	750 psig -320 to 160°F	5 scim of He at -320°F	10,000
5. Rocketdyne	Bipropellant 1.380 dia	800 psig -320 to 160°F	5 scim at -320°F	2,500
Conical Poppet Seats				
1. Rocketdyne	Pressure Regulator	500 psig -30 to 120°F	3 scc/hr of N_2	5,000
2. Benbow	Pressure Relief	500 psig -65 to 160°F	2 ccm of Hyd Oil	50,000
3. Rocketdyne	Three-Way Pressure Actuated	1000 psig -30 to 200°F	10 ccm of RP-1	10,000
4. Rocketdyne	1/4 in. Solenoid, Axial Flow	325 psig -65 to 160°F	0.02 sccm of N_2	10,000
5. Honeywell	Solenoid	300 psia -65 to 165°F	1.0 scc/hr N_2 at 200 psi	80,000
6. Rocketdyne	4 in. Tank Vent (KEL-F seat)	60 psig -300 to 160°F	200 scim He at -290°F	5,000
Spherical Poppet Seats				
1. NAI	3/8 in. Solenoid (17-4 PH against 17-4 PH seat)	5000 psig -423 to 160°F	10 scim of He	10,000
2. Frebank	Pressure Relief	100 psig -450 to 300°F	5 sccm of He	50,000
3. Honeywell	Two-Way Solenoid	300 psia -65 to 165°F	1.0 scc/hr of N_2 at 100 psig	500,000 (100,000 proven)
4. Rocketdyne	1/4 in. Solenoid, Two-Way UDMH	280 psig -20 to 160°F	0.5 sccm of He	50,000
5. Randall Engr.	1/4 in. Hyd Check	3000 psig -300 to 480°F	Zero Bubble of N_2, 5 Minutes	10,000
6. Clary Dynamics	1/4 in. Solenod Two-Way	200 psig	10 ccm of Hyd	2,000,000
7. Rocketdyne	Helium Pressure Relief	4000 psig -320 to 140°F	10 scim of He	1,000
8. Aqualite	Helium Pressure Relief with Integral Burst Disc	4650 psig 30 to 160°F (Teflon-440 CRES Seal)	20 scim of He	1,000

The estimated minimum lives of both pressure regulator and solenoid valves are approximately the same--500,000 cycles. Essentially the same minimum lives are to be expected because the valve seats are the life-limiting factors in both cases. It is believed that pressure-regulator valves for aerospace applications can be developed that have cyclic life in excess of one million because (1) solenoid valve life in excess of one million cycles is possible, (2) cyclic life of some pressure regulators in ground applications has exceeded millions of cycles, and (3) the life-limiting factors of solenoid and pressure regulator valves are essentially the same.

The estimated lives of pressure regulators were statistically less significant than those estimated for solenoid valves. Therefore, it is suggested that all prospective suppliers of future pressure regulators be very carefully screened as to their knowledge and experience with pressure regulator usage in space applications. In summary, solenoid valves and pressure regulators for long-life missions such as Shuttle are feasible if the materials in contact with the fluid or gas medium are completely compatible for these long durations. Accelerated life tests of material/environment compatibility need to be accomplished and provide affirmative results before one can be certain that a particular valve design can survive a 10-year space environment.

e. *Application Guidelines* - Inflight maintenance developments exist for both plug-in valve modules and screw-in assembly modules. These units have been tested successfully on 300 psig systems.

The use of permanent tubing connections is preferred to reconnectable mechanical joints to minimize external leakage. Not all the mechanical joints in a gas or fluid system can be of the permanent type. It is recommended that line replaceable units (LRUs) be defined consisting of logical modular groups of components assembled as a unit. The modular groups of components can be assembled using permanent mechanical joints; then, if a component fails or is operating out of limits, a whole modular group can be replaced using separable mechanical joints. This modular replacement concept would reduce the number of separable mechanical joints in a system and still be consistent with future manned missions.

D. TEST METHODOLOGY AND REQUIREMENTS

1. Development and Qualification Testing

The survey of manufacturers indicated that testing for specific failure modes is not normal practice. However, if prior analysis, such as the FMEA's, indicate a peculiar type of failure mode, they may try to duplicate this failure during the development tests.

Margin tests were performed mainly for over pressure conditions; however, other parameters subjected to margin tests are temperature, vibration and closing forces.

It is recommended that contamination tests be conducted on critical components to determine the level of contaminant that the component can tolerate. For example, a contamination test (Ref 12) was conducted on 17 components that were determined to be sensitive to contamination. Contamination, in excess of the recommended cleanliness levels, was purposely injected into the system to measure component performance in a highly contaminated system. All the components except the pressure regulator operated properly throughout all tests with no degradation caused by contamination. The pressure regulator (1000 psig of N_2 inlet pressure and the outlet pressure setting of 120 psig) failed. The regulator was highly sensitive to contamination due to a small (0.161 in. diameter) poppet seat. The failure was caused by erosion of the main poppet seat as a result of the high velocities across the small seat area. It was recommended that dual-element filters be placed upstream of the pressure regulator. It was further recommended that a single cleanliness specification be referenced on all component specifications, system engineering, and applicable procedures.

2. Accelerated Life Tests

Most valve specialists believe present accelerated testing methods are unreliable, but utilize them since more valid techniques are not available.

Accelerated testing of valves is generally accomplished in several phases, and not by a single test. For example, the cycle life of the valve is commonly verified by using an increased cycle rate. Although this test is very widely used, it does not cope with time-dependent phenomena which may be present such as creep, corrosion,

metal diffusions, etc. For long life valves, it is desirable, but not always feasible, to design out or minimize the time-dependent failure mechanisms. When this is accomplished, the increased cycle rate test yields a valid cycle life projection. The cycling rate must be constrained to prevent secondary effects such as local depletion of lubrication, local hot spots due to friction, or overheating due to power of the solenoid.

When time-dependent phenomena are present, these must be identified and subjected to the appropriate test evaluations, which in many cases, can be accelerated by increasing the temperature. A classic example of an aging mechanism that is somewhat independent of cycle life (depending on the specific valve) is the cold flow or creep that may occur during long dormant periods with a Teflon seat or poppet. It has been demonstrated that Teflon seats with unit stresses to 4000 psi can be designed for long life, with very astute seat design. However, an accelerated high temperature test is needed to verify the absence of cold flow. Empirical data from the Martin Marietta Aerospace indicates that valve seats under normal load for 2 weeks at 150° C should demonstrate a 10 year capability for valves operating at or below room temperature, but experimental data is required to validate this.

Another example of an aging mechanism is the potential bonding of the seat and poppet when subjected to high bearing stress for long periods of time. This problem is addressed in a current study by TRW for JPL (Ref 13) is entitled "Study of Advanced Techniques for determining the Long Term Performance of Components". TRW conducted diffusion tests in which copper and chromium plated Inconel, representing the seat and poppet materials, were clamped together with a load of about 33,000 psi and subjected to 500°C for times of 10, 200, and 400 hours. Bonding did not occur, and since it was estimated that the acceleration factor of the 500°C test was about 100, this couple would not experience bonding for at least 5 years. Although bonding did not occur, mechanical adhesion did occur, due to deformation of the copper, causing it to be "locked in" on the uneven surface of the chromium. TRW concluded that this technique appeared valid for evaluating metal seat-poppet couples, and plans additional work in this area. It was recommended that corrosion tests should also be accomplished before and after diffusion to establish if any net material transfer affects the extent of corrosion or its mechanism.

Yet another example of aging and environmental mechanisms is in the valve solenoid. Long term stability of the potting compound needs to be verified by materials testing methods such as the Thermogravimetric analysis (TGA) technique. Accelerated temperature cycling should then be conducted on the potted solenoid to establish compatibility between the thermal expansion characteristics of the potting and the small wires which are more subject to failure with hard, non resilient, potting compounds.

Martin Marietta Aerospace personnel are predicting long term behavior of the mechanical properties of polymers and composites based on short term tests. Of interest are tests at high temperatures, where kinetics of polymer degradation occurs at an accelerated rate. An effective short-time testing method is thermogravimetric analysis. This technique requires only two hours for data acquisition. A very small sample of polymer is taken through total decomposition by increasing the temperature by a predetermined rate, say 10°C/minute. The results of investigations reported in Ref 14 show that the TGA method does adequately predict the kinetics of large samples at normal use temperatures..

S. H. Kalayan, et al. in the JPL Quarterly article, "Long Term Aging of Elastomers" (Ref 15) relate that analysis of kinetic data obtained indicate that the fluorosilicone rubber will show negligible chemical degradation at room temperature (20 to 25°C) for very long periods, i.e., several decades.

TRW (Ref 13) also reports on a study of acoustic signatures. This subject is also under investigation by GE (Ref 16). The transmittance of acoustic energy through a component provides a sensitive method of detecting changes or anomalies in the discontinuities within a valve. A change in the signature can be an indication of a degradation or aging process or an indication of a defect. This technique is not in itself an accelerated test method, but it is an instrumentation method with the potential of detecting and measuring some aging phenomena. In some cases the trend data, could probably be extrapolated to provide an end-of-life prediction.

3. <u>Life Tests</u>

Nine Titan III Transtage attitude control valves (Moog, Inc., series redundant, monopropellant, Model No. 50-315A) are presently being subjected to a 5-year storage test to determine performance degradation versus storage time (Ref 17). Periodically during this storage phase, tests are conducted including: visual inspection, pull-in/dropout current, response, and leakage. A

sensitivity of each valve's pull-in current requirements to external magnetic factors is observed. The valve, as evidenced by the test results, has performed within specifications. This program will continue through August 1973 at which time a final report will be published covering the entire storage test program.

Sufficient historical data is not yet available to confirm a 10-year service life in the space environment. It is recommended that data be accumulated as available from existing programs for the specific purpose of evaluating life trends.

4. Screening and Wear-In

The survey of manufacturers and users indicated that a modified wear-in is run on the pressure regulators in which the unit is cycled 50 times and adjustments are allowed to assure the regulator is operating within limits. The unit is then cycled another 50 cycles to verify it is regulating pressure within limits. Another wear-in consisted of 5000 cycles where calibration was allowed after each 1000 cycles.

A record tag is maintained on each unit to verify the regulator is functioning within limits. The regulator is calibrated at regular intervals, say 100 cycles. It is recommended that calibration logs be kept on pressure regulator assemblies to determine if the unit is drifting out of limits prior to service usage. Normally the wear-in for both solenoid valve and pressure regulators ranges from 50-100 cycles.

5. Apollo Helium Pressure Regulator Test History

The initial design of the regulator assembly included two dual stage regulating units installed in parallel for the gas pressurization system (normally active unit and the alternate unit). Outlet pressure regulation requirements for the primary stages were 168 ± 4 psig and 181 ± 4 psig (secondary stage) at helium flow rates of 6 to 9 lb/min. and at inlet pressures varying from 4500 to 350 psig.

Qualification testing for Block I was performed during 1956. Although the two units met the qualification test requirements, the primary stage of the regulator leaked excessively after endurance cycle tests. The regulator internal leakage specification limit of 80 scc/min. was satisfied, however, only because leakage through the redundant secondary stage was less than the required limit.

Off-limit testing consisting of vibration, endurance and rupture burst tests, was completed in early 1966. The regulator complied with all performance requirements except during the endurance test. The repeated high energy pressure impacts of a quick opening facility solenoid valve just upstream of the regulator loosened a screw on the surge arrester piston, and the screw fragmented and caused extensive internal regulator damage. The unit was refurbished and testing was continued, but regulator internal leakage was excessive. The Block I regulator was certified June 1966.

The Block II regulator was modified by the addition of a filter at the inlet port, the elimination of metering orifice sizing pins and reduced orifice diameters. The regulating tolerance band was increased to ±7 psi. This change was necessary because Block I testing showed that the regulator could not operate within ±4 psi tolerance. Engine performance analysis showed that the increased tolerance band was satisfactory for the Apollo mission and the Block II procurement specification was revised to incorporate the ±7 psi pressure regulation tolerance. Block II qualification testing was completed in 1966 and the unit was certified in 1968.

6. Failure Mode Detection

The response rate and drift in the response rate of all pressure regulators should be measured before flight. If the response rate exhibits a definite decreasing trend, then the regulator should be considered suspect and replaced.

Similarly trend monitoring of the response rate of solenoid valves should be accomplished to predict the onset of failure. For a single, long duration mission, it will be necessary to conduct repetitive response rate tests prior to flight to obtain the trend. For the Shuttle missions, the response rate could be recorded prior to each use, and the trend established from flight to flight.

E. PROCESS CONTROL REQUIREMENTS

The contamination control requirements and parts inspection covered in the check valve chapter apply also to solenoid valves and pressure regulators. To reiterate, parts should receive ultrasonic cleaning and assembly in either clean rooms or on laminar flow benches. The cleanliness requirements should include the components, fluids, and systems. The clean room specification should be such that contaminant size is less than that specified in the valve design specification as being acceptable. The minimum clean room requirements should be specified in the procurement specification. The Skylab program required that system cleanliness must extend across interfaces with other contractors.

Customers requirements for critical processes should be called out in the procurement specification. Most manufacturers contacted in the survey stated that plating, heat treat, and rubber products require strict process control.

F. PARTS USAGE CONSTRAINTS

The particular designs which are suspect for long life applications are described below.

Gate and tapered plug types of solenoid valves are susceptible to sticking if not properly designed. The slide in gate valves must be thick enough to prevent warpage and hence high frictional forces are introduced under loads. Warpage permits galling and subsequent contamination and leakage. The required torque for tapered plugs is a function of pressure differential and the valve can "lock-up" if not correctly designed. Tapered plug valves are not recommended for general aerospace applications. Avoid designs with plunger seals which would allow leakage into the solenoid valves. Long-duration space missions present a few additional environmental considerations for the valve designer beyond those considerations for earth environment application.

The effects of high vacuum on the sublimation of most materials employed in solenoid valves is not considered a serious structural problem. However, because metals (including plating) that sublime from a warm surface could condense on a cooler surface creating electrical shorts, etc, materials having a high sublimation rate should be avoided. The sublimation rate of material varies directly as the equilibrium vapor pressure for that material. Cadmium plating is a poor material to use in a space vacuum because it has a high vapor pressure. For very long space mission, electrical insulation and thermal protective coatings could suffer radiation damage if radiation exposure is not considered in material selection and design. References 18 and 19 present data to assist the engineer in material selection for space environments. Careful design and material selection should provide the necessary structural integrity and minimize any structural life-limiting problems.

G. REFERENCES AND BIBLIOGRAPHY

REFERENCE

1. *Aerospace Fluid Component Designers Handbook,* Contracts AF04 (611)-8385 and AF04(611)-11316, March 1967.

2. Dr. R. L. Anderson, *et al*: "High Stressed Teflon Seals," *Design News,* October 12, 1970.

3. *Test Adequacy Survey, Phase II, Final Report.* Martin Marietta Corporation, Denver, Colorado, October 1966.

4. "Monitoring Solenoid Valve Position." *Instrumentation Technology,* Lewis Research Center, Page 67, June 1970.

5. *Advanced Spacecraft Valve Technology Compilation.* TRW Systems, NASA Contract NAS7-717, July 1970.

6. L H. Gillespie, D. O. Saxton, and F. M. Chapman: "New Design Data For Teflon." *Machine Design,* February 1960.

7. J. Tracey O'Rourke: "You Can Tailor the Properties of Teflon with Fillers." *The Journal of Teflon.*

8. *Advanced Spacecraft Valve Technology.* Final Report 12411-6011-R000, TRW Systems, NASA Contract NAS7-717, July 1970.

9. Richard J. Salvinski: *Advanced Space Technology for Valves.* ASME Paper 66-MD-61.

10. Ferguson, Tellier, and Mannes: *Valve Design, An Industry Survey.* R-3819-3. Rocketdyne, September 1962 (RL 49182).

11. R. W. Burrows: *The Long-Life Spacecraft Problem.* Paper No. 67-880 AIAA 4th Annual Meeting and Technology Display, October 23-27, 1967.

12. *Study for Cleanliness Level Requirements for Pneumatic and Hydraulic Components - Service Arm Systems, Complex 39.* MCR-69-484, Martin Marietta Corporation, NASA Contract NAS10-5935, September 1969.

13. *Study of Advanced Techniques for Determining the Long Term Performance of Components*. Report No. 16568-6014-R000, TRW System Group, March 1972.

14. H. A. Papazian: *Prediction for Polymer Degradation Kinetics Near Normal Use Temperatures from High Temperature TGA Measurements*. TOS 48634-003, Martin Marietta Corporation, December 1971.

15. S. H. Kalayan, A. A. Mazzeo and R. H. Silva: "Long-Term Aging of Elastomers: Chemical Stress Relaxation fo Fluorosilicone Rubber and Other Studies." *JPL Quarterly Technical Review*, October 1971.

16. Sturglon, L. F.: "Signature Analysis for Mechanical Devices." Vol 2. *Proceedings of the Symposium on Long Life Hardware for Space*, March 17-19, 1969.

17. G. J. Gunderson, Sgt. USAF: *Transtage ACS Valve Storage Tests*. AFRPL-TR-70-150, Project 624A-00DRU, Air Force Rocket Propulsion Laboratory, Edwards, California, January 1971.

18. *Space Materials Handbook, Second Edition*. ML-TDR-60-40. USAF Materials Laboratory Research and Technology Division, Air Force Service Command, Wright-Patterson AFB.

19. *Space Material Handbook Supplement 1 to Second Edition Space Material Experience*. NASA SP-3205, MIL-TDR-64-40, Supplement 1.

BIBLIOGRAPHY

1. *Advanced Technology Studies.* SD71-267, North American Rockwell NASA Contract NAS7-200, December 1971.

2. *Handbook of Long-Life Space Vehicle Investigation.* M-70-16, Martin Marietta Corporation, Denver, Colorado, December 1970.

3. *Apollo Service Propulsion System Rocket Engine Bipropellant Valve Improvement Program.* Report 8317-P1, Aerojet Liquid Rocket Company, NASA Contract NAS9-8317, July 1970.

4. W. D. English, H. D. Samuel, Jr: *Failure Mechanism of Attitude Control System Valve in Chlorine Pentafluoride Service.* Report AFML-TR-71-94, McDonnell Douglas Astronautics Company, USAF Contract F33615-70-F-1760, July 1971.

5. G. F. Tellier and T. R. Spring: *Contaminant Particles in Metal-to-Metal Closures.* Report AFRPL-TR-71-112, USAF Contract F04611-70-C-0048, December 1971.

6. *Aerospace Mechanisms.* Edited by G. G. Herzl, Lockheed Missiles and Space Company, Distributed by J. W. Stacey, Inc., 1970.

7. J. C. DuBuisson: *Long Life Reliability Problems and Solutions for Valves under Spatial Environments.* Paper No. VII H.1, Joint National Meeting American Astronautical Society and Operations Research Society, June 17-20, 1969.

8. L. R. Bell, Jr.: *Long Life Reaction Control System Engine and Valves.* Paper No. 70-603, AIAA 6th Propulsion Joint Specialists Conference, June 15-19, 1970.

9. *Parts, Materials, and Processes Experience Summary.* CR114391, Lockheed Missiles & Space Co. Inc., NASA Ames Research Center, February 1972.

10. *TOS 860 Long-Life Components: Final Report for 1967.* Martin Marietta Corporation, Denver, Colorado.

11. G. F. Tellier and J. W. Lewellen: *Poppet and Seat Criteria for Contaminant-Particle Resistance.* Report AFRPL-TR-70-1, April 1970.

PERSONS CONTACTED

1. Mr. Morley Friedell, A. Wright, Martin Marietta Corporation, Denver, Colorado, Dept. 0431

2. Mr. Jack Schieder, Carleton Controls, New York (716) 652-8100.

3. Mr. Stan Kawalski, Marotta Valve Corp., Boonton, New Jersey, (201) 334-7800.

4. Mr. Ron Book, Moog Valve, E. Aurora, New York (716) 652-2000.

5. Mr. Stan Summers, Calmec Mfg. Corp., Los Angeles, California (213) 771-4100.

6. Mr. Joe Pauly, Sterer Eng. and Mfg. Co., Los Angeles, California, (213) 245-7161.

7. Mr. Bruce Waterman, Futurecraft Corp., Los Angeles, California, (213) 330-1611.

8. Mr. Vince Butler, Parker Hannifin Corp., Los Angeles, California (714) 833-3000.

9. Mr. Frank Green, Garrett AiResearch Corp., Los Angeles, California (213) 323-9500.

10. Mr. Dick Schuman, Janitrol Aero., Columbus, Ohio. (614) 276-3561.

11. Mr. Bob Hamilton, Robertshaw Fulton, Los Angeles, California (714) 535-8151.

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X. THERMAL CONTROL VALVES

by P. J. Powell

X. THERMAL CONTROL VALVES

A. INTRODUCTION

Thermal control valves are installed in a coolant system for the purpose of regulating coolant flow through a heat exchanger to maintain required atmospheric or equipment temperature. Thermal control valves may be used to control fluid flow through an external space radiator to maintain a constant temperature at the radiator outlet. An example of this application is depicted in Figure 1. They can also be used internally in the space vehicle to divide the coolant flow between equipment and atmosphere heat exchangers. In this application, the electronic equipment is adequately cooled, and the space vehicle atmospheric temperature can be controlled using the heat generated from the electronic equipment.

1. Valve Types

Thermal control valves for the above applications are normally three-way (three ports) and can be categorized into two general types. The Type A design incorporates the temperature sensor and control mechanism within the valve housing, and valve power element operation is similar to that of a thermostat valve for an automobile engine. This type is completely mechanical, and requires no external temperature sensor or electrical power.

The valve cylinder is positioned by forces generated within an internal power element containing a wax-type material with a high ceofficient of expansion. As the discharge coolant temperature changes, the wax-type material expands or contracts, repositioning the valve cylinder to maintain a relatively constant discharge temperature (see Figure 2 for a hysteresis curve of the sensing element which shows the relatively short piston displacement).

An attractive feature of this type of valve is that it eliminates the need for a separate external sensor and controller/actuator. This allows a high system reliability and reduces overall system weight. Some disadvantages of this type of valve are: (1) temperature control accuracy is limited to a range of approximately ±2.8°C; (2) response time is relatively slow; and (3) the valve design does not lend itself to simple adjustment or change in control point. A failure of the sensor or positioning mechanism would require system shutdown and valve disassembly for repair. An example of a Type A valve is illustrated in Figure 3.

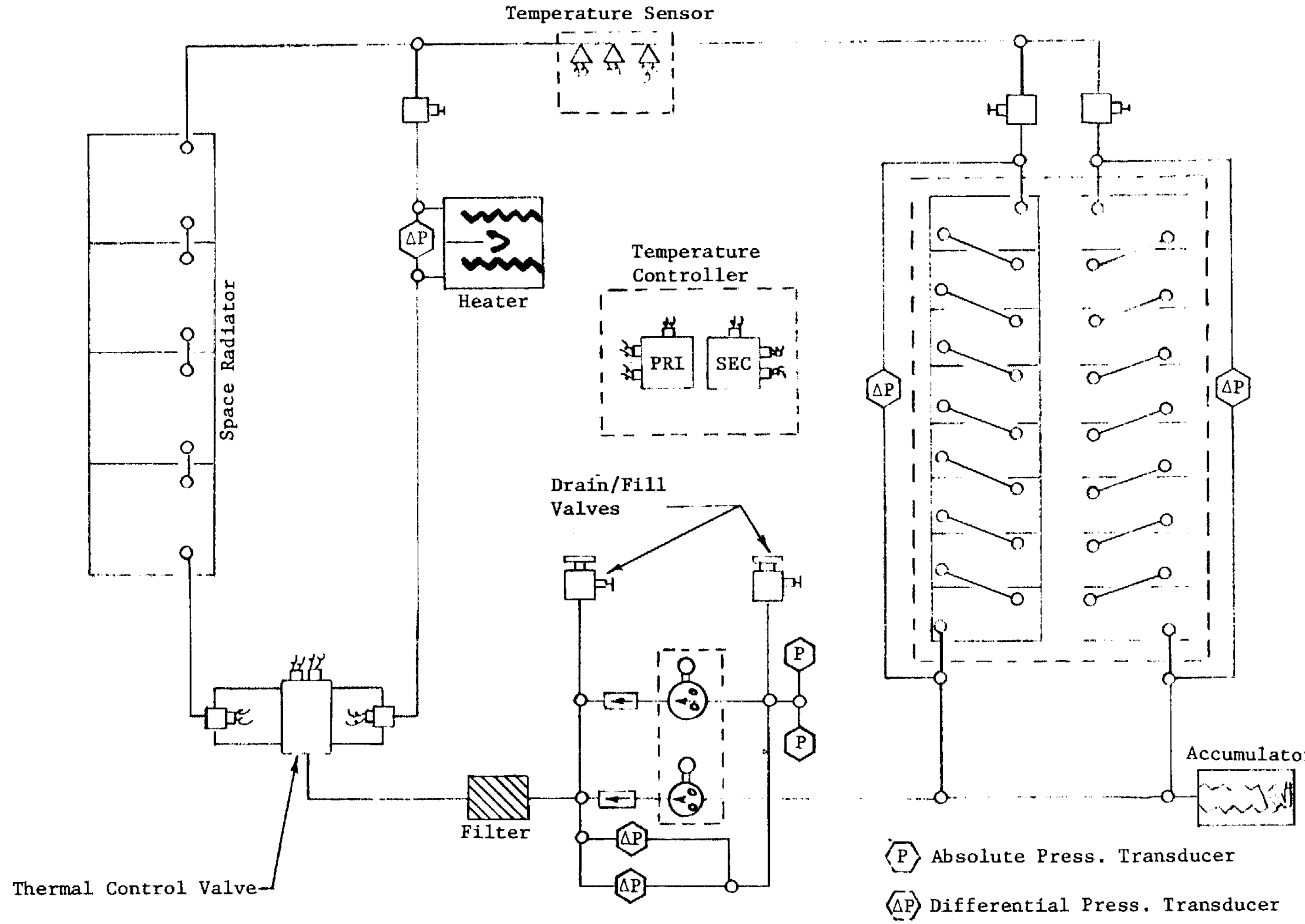


Figure 1 Schematic Diagram, Thermal Control System, Apollo Telescope Mount

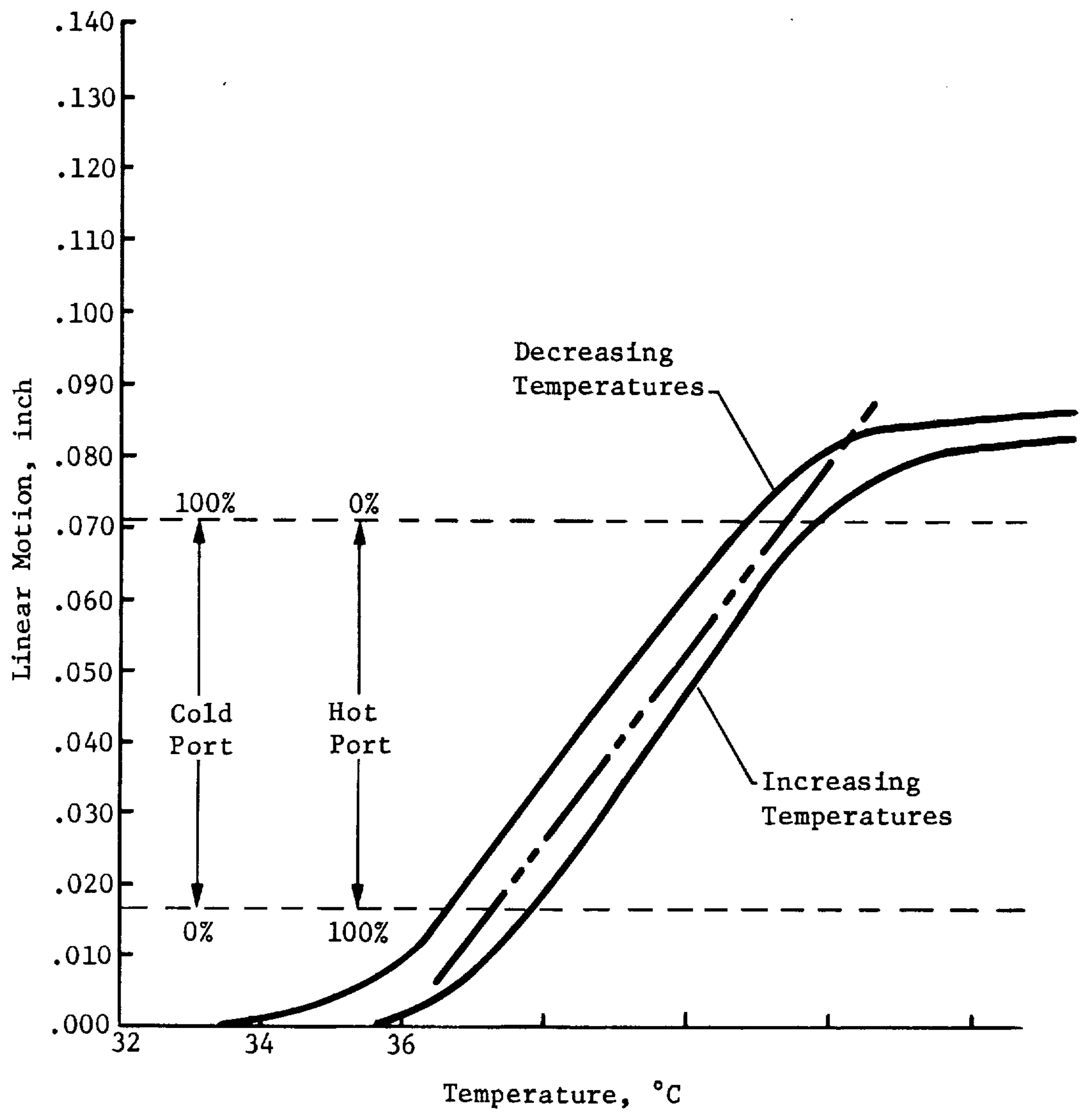


Figure 2 Typical Thermostat Hysteresis Curve

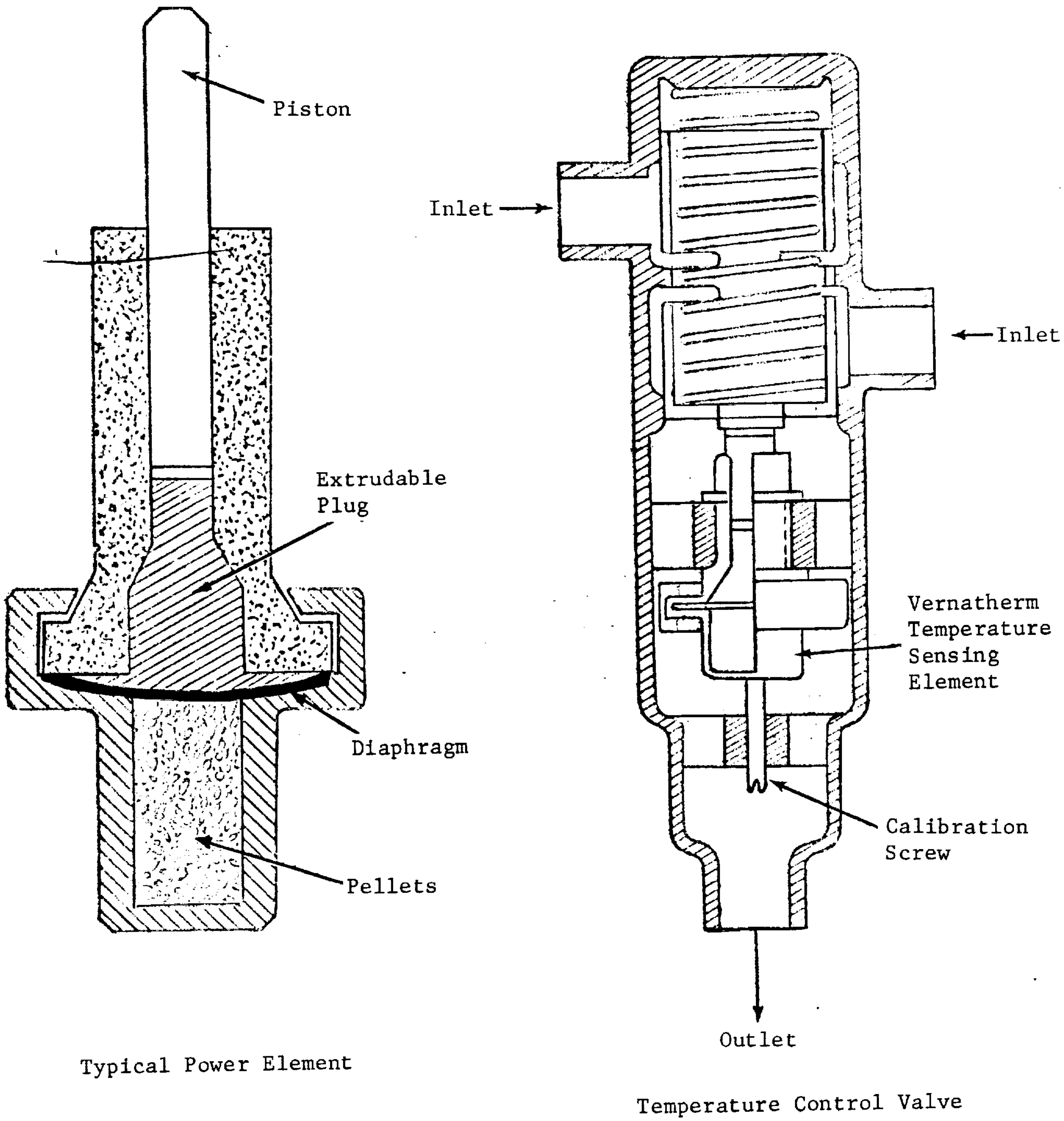


Figure 3 Type A, TCV

The Type B design is a valve that requires positioning of the valve stem by an external electronic controller/actuator assembly, and also requires that a separate temperature sensor be installed in the piping at the point where temperature control is desired. The sensor, normally a resistance element, provides the intelligence to the valve controller/actuator. The valve is positioned to satisfy the sensor and accomplish temperature control. The Type B design is capable of controlling to ±.5°C or better. It can be readily adjusted through a wide range of set points and offers fast response time. Another advantage of this design is that the controller/actuator and sensor can be readily replaced should a failure occur, without interrupting coolant flow. Disadvantages of the Type B design in comparison with Type A are reduced reliability of the overall system (including controller and sensor) an increased system weight. An example of a Type B valve is illustrated in Figure 4.

B. GUIDELINES FOR LONG-LIFE ASSURANCE

Service life estimates for thermal control valves (TCV's) used in fluid systems with internal temperature sensing ranged from 5000 hours to three years and from 8000 cycles to 20,000 cycles. The cycle life of TCV's used in fluid systems with external temperature sensing is 15,000 cycles (by test) and a three to five-year service life is obtainable. The TCV service life requirements cannot be met for long duration programs unless maintenance and restoration is permitted.

Listed in the order of most probable occurrence, the TCV failure modes are:

1) Failure to modulate flow within limits;

2) External leakage.

1. Design Guidelines

Due to the relatively short service life of current TCV designs it is suggested that:

1) TCVs be designed for both manual or automatic operation whether the temperature is sensed internally or externally, to provide additional flexibility in manned space vehicles;

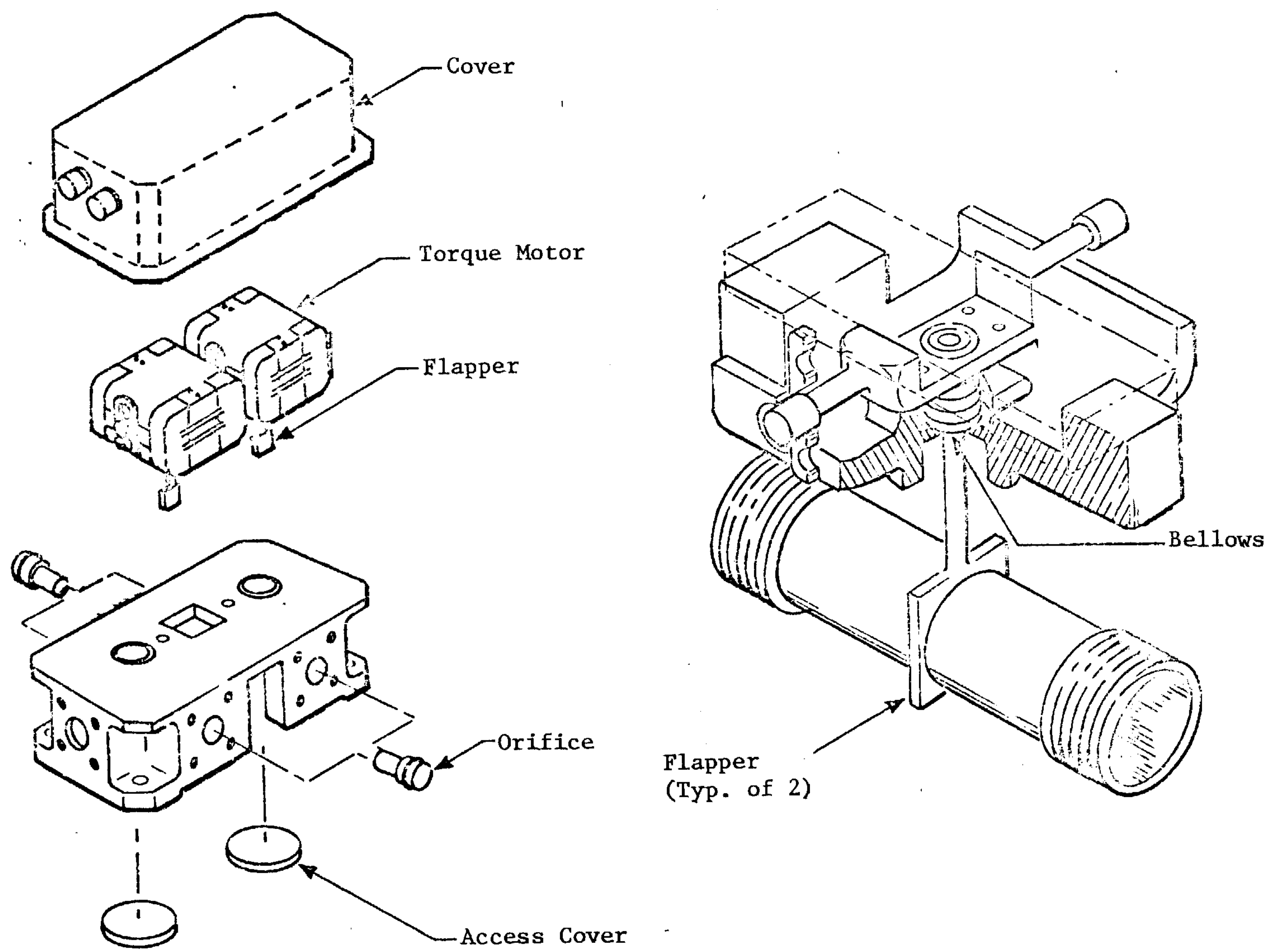


Figure 4 Modulating Flow Control Valve (Thermal Control Valve)

2) The designer consider modular replacement ease;

3) Standby redundancy be used when the life limiting failure mode for a TCV is wearout.

4) Design large flow paths (areas) through thermal control valves and main sealing surfaces to reduce flow velocities and erosion of seats.

5) Configure housings (as far as possible) to eliminate areas that can entrap contaminants. Use a "swept by" design where flow will tend to pass contaminants through the component.

6) Insure that materials are compatible with the working fluid including maximum expected fluid impurities. The evaluation must consider dynamic as-well-as static applications and must consider temperature, pressure, and phase variations of the fluid.

7) Minimize component induced contamination:

   a) Material selection should consider the effect of wear particle size on useful life. Particle sizes vary as the Young's Modulus divided by the square of the compressive yield stress;

   b) Use rolled threads in preference to machined threads to minimize burrs and achieve higher strength;

   c) Minimize dead-end passages and capillary size passages.

8) Leaks due to contamination are minimized if the seat loads are sufficient to plastically yield a trapped contaminant particle on the sealing surface. System contaminants must be identified to determine particle size and material properties.

9) Avoid sliding parts; they not only induce contamination, but entrap contamination.

10) Develop valving elements with quick opening areas to preclude chattering, instability and high seat velocities.

11) Spherical or tapered poppets are recommended. These poppets are self-aligning, but require precise alignment of the seat and the poppet assembly.

12) Insure guiding of poppet onto seat to allow for maximum alignment and eccentricity tolerance (i.e., use a large length/diameter bearing surface to guide ratio - minimum 2:1).

13) High safety factors should be used in conjunction with high seat loads since low seat stresses are desirable for long life.

14) Lipseals supported with a back-up ring to resist undesirable influences of pressure loading are superior to designs without this provision. It is recommended that lipseals guiding piston rods or poppets use back-up sealing rings in addition to the main lipseal.

15) To minimize transient stresses due to poppet travel, provide positive stops at the end of travel.

16) Seal retention methods shall prevent seal distortion, creep or dislodgement.

17) Reduced life due to vibration sensitivity is minimized by decreasing available clearances in bearings and guides, avoiding large overhung moments, and restraining lateral motion of poppets.

18) Control stress corrosion by avoiding stress corrosion susceptible materials or design parts to operate at low stress levels.

19) Justify the use of dynamic seals. They are subject to wear and cause unpredictable drag.

20) Control external leakage by:

    a) Requiring welded valve body construction;

    b) Requiring vacuum melt bar stock;

    c) Impregnating valve casting with sealant.

21) The solenoid open and close force margin goal is 300%. Failure of the poppet to open (or close) may be caused by binding of the plunger or insufficient solenoid force.

22) Prevent insulation deterioration and subsequent solenoid shorts that can lead to reduction in electromagnetic force by:

a) Coating solenoid valve lead wires with an abrasive resistance covering;

b) Taking special precautions when joining the lead wire to the solenoid coil wire;

c) Potting coils and lead wires to prevent movement during vibration and operation;

d) Using wire gages less than 40.

2. Process and Control Guidelines

1) Ultrasonically clean valve parts; assemble in specified level-clean areas, and govern by a single contamination control specification during test. This same control specification should also govern the test fluid media used.

2) Use a fabrication barrier (bag) to protect clean parts. Consider using nylon or polyethylene to prevent creation of contamination due to chaffing of the barrier by the parts.

3) Vendor controls shall guarantee that the valve contamination particle size and count will not exceed specified limits. Documentation is required.

3. Test Guidelines

1) Conduct contamination susceptibility tests during development to determine the level of contaminant that the valve can tolerate.

2) Perform 50 to 100 run-in cycles to eliminate components with latent manufacturing defects.

3) Do not rapid cycle valves for functional verification in a dry condition because the lack of fluid damping can increase seat stress and reduce life.

4) Conduct life cycle endurance tests under operational conditions. For long-life applications, the valve cycle life parameter must be known.

5) Test for corrosion of materials in the presence of the working fluid and the maximum expected impurities expected during operational life. Evaluate pitting and stress corrosion as well as penetration rates over a large surface area.

6) Consider use of holographic terferometry test methods for fluid compatibility evaluations. These methods allow evaluation of the onset and time variation of the corrosion process and permit three dimensional evaluation of localized effects.

7) After operational valve use, measure the response characteristics of the valve and perform trend analysis to identify wear trends.

8) Perform leak checks after valve use and subject data to trend analysis.

4. Application Guidelines

1) To avoid contamination and to prevent leakage, install components into systems using permanent connections, such as welded, brazed, or swaged connections rather than reconnectable joints.

2) The controller/actuator of a Type B valve can be readily replaced without interrupting coolant flow.

3) System design should identify the Line Replaceable Units (LRUs) consisting of individual valves or groups of valves for ease of remove/replace activities in the event of component malfunction.

4) A solution of long-life and reliability problems is the application of valves into standby redundant configuration.

C. LIFE LIMITING PROBLEMS AND SOLUTIONS

1. Failure Mechanisms Analysis (FMA)

Summarizing from Table 1, the failure modes of the TCV in order of most probable frequency are:

1) Valve instability or internal leakage causing failure to modulate flow within limits, and;

2) External leakage.

a. Failure to Modulate Flow Within Limits - The prevalent life-limiting factor of the internal temperature control valve mechanism is the dynamic seal that prevents leakage around either the valve stem or other mechanical linkages between the valving element and the actuating device.

Studies of the Skylab Airlock Module valves, showed that leakage around the valve stem would be lost from the system and would eventually cause system failure. Inadequate sealing of the valve used on LM or Titan IIIC would cause excessive internal bypass of coolant between the temperature sensing portion of the valve and the flow regulating portion of the valve. This would degrade the system capability to control temperature.

Recommended solutions to the seal leakage problems are:

1) Develop and test a packless valve that uses a metal bellows around the valve stem instead of conventional valve packing;

2) Develop and test superior seal materials;

3) Design a system that will allow convenient replacement of failed parts in the space environment without compromising performance.

Valves being manufactured for the Skylab Apollo Telescope Mount (ATM) coolant system by LTV already incorporate the metal bellows, packless valve concept. These valves have been tested under earth environmental conditions through 15,000 cycles, or an approximate equivalent of 2 years operating time. Valve bellows have been successfully tested through 10×10^6 cycles.

Table 1 Failure Mechanism - Thermal Control Valves

Part and Function	Failure Mode	Rel. Rank	Failure Mechanism	Detection Method	How to Eliminate/ Minimize Failure Mode
Flapper Assembly - (proportions flow through valve) or Internal sensing element and control mechanism - (Proportions flow through valve)	Failure to modulate caused by internal leakage or valve instability.	1	Internal leakage could be due to broken or deformed torque bar or flapper, binding of torque bar pivot, internal leakage of bellows, or irreular flapper seat or leakage past dynamic seals. Instability was due to locating sensor away from valve control. External leakage through seals or valve housing.	<u>Pre-Installation</u> The internally sensed TCV's sensing elements leakage and displacement can be verified prior to system usage. <u>Post-Installation</u> The externally sensed TCV flapper positions can be monitored by T/M or on-board display. The internally sensed TCV's are monitored by system parameters.	Develop and test a packless valve that uses metal bellows around the valve stem. For manned systems provide manual overriding controls without compromising reliability during normal automatic control mode.
Valve body - (retain valving elements and contain flow media)	External leakage.	2	External leakage through seals or valve housing.	<u>Pre-Installation</u> The proof pressure tests reveal this type of failure. <u>Post-Installation</u> External leakage can be detected visually for manned flights. T/M can be used to detect system problems on unmanned flights.	Provide welded valve housing construction or redundant static seals.

Mr. J. Potter (MSFC) stated the primary problem with the Type A TCV is leakage and contamination resulting in system temperature out of limits. The primary problem with the Type B TCV is instability because the sensor is located away from the valve element.

External leakage occurs due to leakage through the valve body static seals. This problem can be eliminated by welding the valve body sections.

No method exists to determine internally sensed TCV status, whether it is functioning or not, except by system instrumentation. Externally sensed TCV valves can be monitored (whether it is open or closed) by position switches.

2. Design

a. Selection Criteria - Table 2 shows a design factor for long life assurance that should be considered in TCV design. The externally sensed TCVs are used to modulate flow control to maintain a cold plate inlet temperature within ±.5°C limits. The internally sensed TCVs are also used to modulate flow control but have a much slower response and hence are used in applications where the temperature tolerances can be greater.

b. Results of Industry Survey - Five companies were contacted that have experience in the design, development, manufacture and test of thermal control valves for space vehicle applications. Comments of personnel of the five companies are summarized in Table 3. The companies contacted were Garrett AiResearch, Janitrol, Hamilton Standard, AC Electronics, and Ling-Temco-Vought (LTV). LTV is providing a TCV with external temperature sensing for the Skylab ATM equipment. Thermal control valves with internal temperature sensing have been furnished for NASA and military programs by the other companies.

The LTV valve is a reed, or "flapper," type three-way valve. It is controlled by a remote sensing element and an electronic controller and is positioned by redundant torque motors. The design of this valve utilizes essentially the same concepts as the TCVs used on the Apollo Block II flights. This type of valve was also recommended for the Shuttle program. It has a stainless steel bellows instead of conventional non-metallic seals to reduce the probability of valve leakage. The LTV representative indicated that the valve had never been tested to failure (no margin testing), so operating life was unknown. Apparently, valves have been tested without failure for 15,000

Table 2 Design Factors for Long-Life Assurance Part/Component: Thermal Control Valves

DESIGN FACTORS	REMARKS
Clearance between sliding parts	Contamination lodged between close fitting parts in the valving mechanism can cause sticking or cocking, areas that can entrap or cause contamination buildup should be eliminated.
Valve actuation force margins	Conservative force margins are necessary to provide sufficient force to control and modulate fluid flow through the valve.
Seat configuration	Fluid flow around the valve seat may cause turbulence and high pressure drop. Metal to metal seats are used in these designs because the leakage requirements are not stringent.
Dynamic seals	Dynamic seals are subject to wear and aging and should be avoided.
Static seals	Brazed or welded joints are preferred to captive types of seals for valve construction. Also non-separable mechanical joints are preferred to separable mechanical joints but the use of modular component assemblies should be investigated.
Materials compatibility	Avoid plating of internal parts exposed to fluid. If polymers are used in valve construction, the effects of the propellants, temperature, vacuum, must be analyzed.

Table 3 Results of Manufacturing/Agency Survey - Thermal Control Valves

QUESTIONS	CONSENSUS ANSWER
1. Has your company been involved in the design, development, manufacture or test of thermal control valves for use in a space vehicle environment? Which vehicles?	Yes, TCV's have been supplied for Mercury, Gemini, Titan III, Apollo, and Skylab Programs. The service lives vary from 7.5 hours to 8 months.
2. What is the expected life of the thermal control valves?	TCV's - internal temp sensing 5000 hr to 3 yrs and from 8000 to 20,000 cycles (estimate). TCV - external temp sensing 3 to 5 yrs service life and up to 15,000 cycles (by test).
3. What failures have occurred? What are the failure causes?	The internal temp sensing TCV experienced a dynamic seal problem which allows the wax to leak and the temp limits to drift. The external temperature sensing TCV has experienced an instability problem because the sensing element is a great distance from the valve.
4. What solutions do you suggest for the above failure modes that would enhance the operational life or increase the probability of success?	Develop a packless valve, eliminate sliding seals, provide a manual override, and join the valve together with a weld.
5. To achieve long life what design features should be incorporated into the thermal control valves for space applications?	Provide for inflight maintenance of failed valves. Redesign the valve by eliminating the sliding contact by: a rubber boot and flexure of the piston or encase the piston with a bellows.
6. How did you determine part life?	Due to the relatively short life requirements - life was based on cycle requirements and test data.
7. Did you test for specific failure modes?	The seal problem was anticipated and one company devised a high temperature test to check seal.
8. Is accelerated testing and margin testing used?	No, however, cycle tests were performed. Skylab refrigeration system is operated 43 to 45 days on ground prior to launch.
9. What process controls are necessary for long life?	Parts are 100% inspected, cleanliness requirements are necessary as noted in the other valve studies, and the piston surface finish is critical. One company would check the valve after cycle tests were performed to determine the variation in displacement of the valve piston, a difference of greater than 7% was cause for rejection.

cycles and bellows have been tested without failure for 10 million cycles. This suggests that the operating life goal of 10 years may be attained with this design, depending on the number of cycles per hour required to maintain system control.

The TCV with internal temperature sensing manufactured by Hamilton Standard for the LM thermal control system used Viton A and silicone seals. These seals were satisfactory for the 48-hour mission operating time. However, the representative contacted believed that sufficient experience and test data was obtained to indicate that significant design changes would be necessary to achieve an operating life goal of 1 to 10 years and that maximum attainable life could only be determined after further design, development, and testing is complete.

The valve used by AC Electronics to control temperatures for the Titan IIIC guidance equipment was required to operate only 7.5 hours in flight. However, qualification testing, operation of non-flight hardware and checkout of flight hardware have resulted in accumulation of an estimated 1000 hours of troublefree operation of a single valve. In addition, two valves have received 500 cycles each and two other valves received 1000 cycles each. Basing his judgment on that experience, the AC Electronics representative stated that design improvements and testing could extend the life of their valve to approximately 3 years.

Garrett AiResearch is presently manufacturing and testing TCVs for the coolant system for the Airlock Module, Multiple Docking Adaptor and Orbital Workshop for Skylab. Company representatives estimated an operating life of 8000 cycles, but information was not available on the number of valve cycles required per hour for system operation. Actual required operating time for the valve is 140 days (3360 hours) out of the total 8-month mission. This indicates that variations in system cooling load, and fluctuations in radiator ambient temperature would require approximately 2.4 equivalent valve cycles/hour to accomplish system performance. The actual operating life of this valve will not be determined until completion of system testing by the McDonnell Douglas Corporation. A Garrett AiResearch representative estimated that a 2-year operating life was attainable, with additional research, development and testing.

In general, operating life requirements for current space vehicle applications have been relatively short, the longest being a 14-day Apollo mission. Maximum design life for the Apollo thermal

control valves is 500 hours, with qualification testing for 1200 hours. The Mercury and Gemini programs also used thermal control valves, but for short operating periods.

c. Alternate Approaches - A trade study was completed by Martin Marietta Aerospace to consider the thermal methods for long life applications up to 10 years. This study (Ref. 1) examined fluid loop, refrigeration system, heat pipes, and passive methods. The study recommended the use of a liquid coolant system similar to those described for Skylab to cool electronic equipment. This system would use thermal control valves to regulate coolant flow through an external radiator to control temperature.

Valve reliability is a major consideration in the design of low thrust systems. Reference 2 describes a flow-control device which, in certain cases, allows one to eliminate mechanical valves from the system. The resulting, "valveless design" has the advantage of high reliability, low leakage rate, and the feature of being fail-safe. The primary disadvantages are response times that are on the order of minutes, larger power requirements than those of the equivalent mechanical valve, and a general limitation to systems of less than approximately 10^{-2} lbf thrust. In many low thrust space applications such as drag makeup, station keeping, orbit adjustment, and satellite inversion where very small thrusts are used for extended periods of time, thrust start-up and termination times on the order of minutes are acceptable.

The principle involved utilizes the phenomenon of solid-vapor phase change associated with subliming substances. In this respect, the valveless design is limited to applications suitable for subliming solid thrusters. Flow control for the valveless design is affected by placing a condensing element at some point in the propellant delivery line. The condensing element can be a screen or porous plug upon which the propellant condenses to terminate flow. Flow is terminated by allowing the screen temperature to fall below the equilibrium value; as a result, the propellant is condensed on the screen and eventually blocks the flow through the pores. Resumption of flow is effected by reversing the process and heating the screen to a temperature above the equilibrium value, thus causing the condensed propellant to vaporize and reopening the pores to flow. Thermal control of the condensing element is maintained by an electrical heater used in conjunction with a heat-rejection device such as an optical solar reflector.

d. Operating Life - The maximum demonstrated TCV life of those surveyed was only 1200 hours. Maximum life demonstration of current NASA programs will be 3360 hours and 8000 cycles for Skylab applications. The estimates of mean life of TCVs with internal temperature sensing varied between 1 and 3 years. It is believed that a 3 year service life can be obtained using state-of-the-art and hi-rel part controls for TCVs with internal temperature sensing.

Cursory life estimates for TCVs with external temperature sensing using metal bellows were as high as 10 years. Manufacturer's representatives had no data to substantiate their estimates. A long life study of solenoid and pressure regulator valves in this report and in Reference 3 indicates that the mean lives of standard aerospace quality valves could be 10 year service life, 500,000 cycles, or both. A TCV using external temperature sensors uses design methodology basic to all valves, seals, and mechanisms. It is believed that the service life of the subject TCVs (Type B) could be 5 years, 100,000 cycles, or both. The lives of hi-rel valves could be longer.

Further advancements in the state-of-the-art must occur before space vehicle thermal control valves with a 10 year life are assured. Extensive testing and possible design improvement of both internally and externally actuated valves must be accomplished before a realistic appraisal of attainable operating life can be established. Also, the alternate approach of accomplishing the required 10-year life by designing the thermal control system for replacement of valves in the operating environment should be explored further.

e. Application Guidelines - Maintainability was not a prime consideration in the design of thermal control systems for previous space programs. Flight duration was relatively short, and the philosophy was to provide assured system performance by designing for high reliability. Apparently this same philosophy is being followed for thermal control system design on the Skylab program, which has a mission duration of approximately 8 months. No provision for restoration or replacement of TCV components on Skylab has been made to ensure continued system performance throughout the mission.

As the aerospace industry plans for the longer missions of up to 10 years, maintainability considerations require greater concern.

Although a theoretical analysis might indicate that a sufficiently reliable TCV can be designed to give trouble-free operation for long periods of time, considerable development effort plus extensive testing in a simulated space environment will be required to provide the necessary confidence that long life has been achieved. An alternate approach is to emphasize maintainability and reduce the scope of development and testing of valve designs.

Some maintainability considerations for design of thermal control valves are:

1) Possible use of quick disconnect fittings on valves or modules to allow fast removal and replacement.

2) Methods to isolate, drain, and refill valves with coolant while performing maintenance in a space environment.

3) Maximum use of manual overriding controls without compromising high reliability during the normal, automatic control mode.

4) Provision for separation of gases trapped in the coolant system during valve replacement.

5) Provision for additional onboard storage space and increased weight capacity to accommodate spare parts.

6) Increased valve complexity and weight to satisfy the modularized, quick removal approach.

D. TEST METHODOLOGY AND REQUIREMENTS

1. Qualification and Accelerated Tests

Qualification tests should be run for all extreme operating conditions. During testing of the LTV Type B valve for the Skylab ATM, the valve failed to maintain the temperature within limits when the radiator fluid outlet temperature was lowered to its minimum expected value. The problem was evidently caused by a flow instability that was corrected by dampening the valve spring force.

Neither of the two valve types have received accelerated testing. Margin tests were performed on pressure, valve force, and endurance cycle tests. Comments on accelerated testing techniques are presented in Chapter IX.

2. Life Tests

The seal used with Type A TCVs is a source of concern for long life applications. One manufacturer exposed the valve to 71°C for one week to verify adequate sealing. During this time period, the valve was operational to maintain the temperature environment. Additional test data is required to verify seal life for long-life applications.

The LTV Type B valve has a cycle life requirement of 7500 cycles. NASA is currently cycling the Skylab system which utilizes this valve until a failure is detected (15,000 cycles have been achieved to date without failure). This valve is being proposed for the Shuttle program.

To date, program requirements have been for relatively short service lives for coolant systems and TCVs. For that reason extensive life cycle tests have not been performed. It is recommended that life cycle tests to failure be performed on both type TCVs.

3. Screening and Run-in

The externally sensed TCV coil of the LTV Type B valve is subjected to a 24 hour run-in.

The Skylab coolant loop is exposed to a functional test of approximately 45 days in the Vertical Assembly Building at KSC. The system is deactivated during launch to reduce pressure loads on the

system during launch to reduce pressure loads on the system during maximum acceleration. The coolant loop is reactivated shortly after launch, and remains on throughout the eight month mission.

Run-in tests on TCVs have been characteristically 10 to 50 cycles.

E. PROCESS CONTROL REQUIREMENTS

A TCV employing external temperature sensors uses construction methodology basic to all valves and the reader is referred to Section E, IX Solenoid valve chapter for Process Control Comments.

The Type A TCVs have experienced problems in the sensing elements. It is recommended that commercial, nonaerospace sources for the thermal sensing element be eliminated or the design altered to reduce internal leakage of the unit.

Improper piston surface finish can cause wear and corrosion thus limiting the life of the TCV. Therefore, the surface finish must be specified in the procurement specification and verified during manufacturing.

F. PARTS USAGE CONSTRAINTS

The TCV life requirements for the Space Shuttle and long duration military space applications cannot be assured with present hardware and thus maintenance and restoration may be required. This study indicates that TCV service life expectancies are 1 to 5 years using present state-of-the-art hardware. The internally sensed TCV (Type A) should not be used for long-life applications unless the design is improved as previously suggested. The externally sensed TCVs (Type B) show promise for long-life applications; however, testing must confirm this assertion.

Viking and Skylab TCV life requirements can be met without maintenance and restoration; however, primary and secondary coolant loops are used to increase the system reliability.

G. REFERENCES AND BIBLIOGRAPHY

1. Personal Contacts

Mr. Frank Green and John Rousseau, Garrett AiResearch, Los Angeles, California (213) 670-0131.

Mr. Don Rethke, Hamilton Standard, E. Hartford, Connecticut, (203) 565-5629.

Mr. John Wean, AC Electronics, Milwaukee, Wisconsin, (414) 762-7000.

Mr. Harry Scott, Ling-Temco-Vought, Dallas, Texas, (214) 262-3222.

Mr. Dick Schuman, Janitrol, Columbus, Ohio (614) 276-3561.

Mr. Jack Potter, MSFC (205) 453-5700.

Mr. Stan Hightower, Dave Rich, Martin Marietta Aerospace.

2. References

1. V. H. Thomas, *Mechanized vs Non-mechanized Thermal System*, Martin Marietta Aerospace, March 1970.
2. W.L. Owens, Jr., "A Flow Control Valve Without Moving Parts," *Aerospace Mechanisms*, Lockheed Missiles and Space Company, Distributed T. W. Stacey, Incorporated, 1970.
3. J. C. DuBusson, *et al.*: *Handbook of Long Life Space Vehicle Investigations*, M-70-16. Martin Marietta Aerospace, Denver, Colorado, December 1970.

3. Bibliography

Advanced Spacecraft Valve Technology, Final Report 12411-6011-R000, NASA Contract NAS7-717, TRW Systems Group, July 1970.

G. H. Pearson: *The Design of Valves and Fittings*. 1964.

L. D. Jaffee and S. B. Rittenhouse: "How Materials Behave in Space." *Materials in Design Engineering*, Vol. 56, No. 3, September 1962.

"How Radiation Affects Engineering Materials." *Materials in Design Engineering*, Vol. 52, No. 1, July 1960.

H. O'Connor and J. Harvey: "Anatomy of a Design Engineering - a New Valve." *Product Engineering*, Vol. 35, No. 20, September 1964.

T. W. Edwards: "Valves." *Power*, Vol. 105, No. 6, June 1961.

"What to Consider When Selecting Valves." *Heating, Piping and Air Conditioning*, Vol. 31, No. 1, January 1959.

XI. PRESSURE VESSELS AND POSITIVE EXPULSION DEVICES

by P. J. Powell

XI. PRESSURE VESSELS AND POSITIVE EXPULSION DEVICES

A. INTRODUCTION

This chapter is addressed to the long-life assurance of tanks, including surface tension devices and conventional expulsion systems. This study concentrates on tanks and expulsion devices because pressure vessels exhibit the same long-life failure modes as tanks. Expulsion devices are designed to provide a continuous fluid flow on demand to spacecraft systems for the duration of the space flight. The applications for the tanks and expulsion devices include: auxiliary propulsion, water, thermal control, and hydraulic systems. The expulsion systems are required to prevent vortexing, gas pull through, slosh, and containment of the bulk medium adjacent to the tank outlets under low or zero gravity conditions.

There are five major types of expulsion devices: (1) bladders, (2) diaphragms bladders, (3) bellows, (4) pistons, and (5) surface tension devices. Though a much newer development than the conventional expulsion devices, the surface tension devices show much promise. Sketches of each of these expulsion devices and tanks are presented and accompanied by a brief explanation of their specific function.

1. Tanks

Tanks which utilize gas stored in a liquid state may require baffles to control the sloshing and swirling of the fluid. An exploded view of one of the LM propellant tank configurations is shown in Figure 1. This figure illustrates typical baffles and tank entry and fitting penetrations. This configuration is an approximately spherical tank of approximately 54 in. diameter fabricated from 6Aℓ-4V titanium alloy.

2. Expulsion Devices

The purpose of expulsion devices is to position a quantity of fluid over the tank outlet under conditions of adverse or zero gravity. The fluid should not breakdown, leak into the gas system, or be contaminated in any way under the influence of the expulsion system. A gas pressurant system provides the motive power for the expulsion. Expulsion devices are employed on several different systems such as the Skylab water system which was a bellows expulsion device.

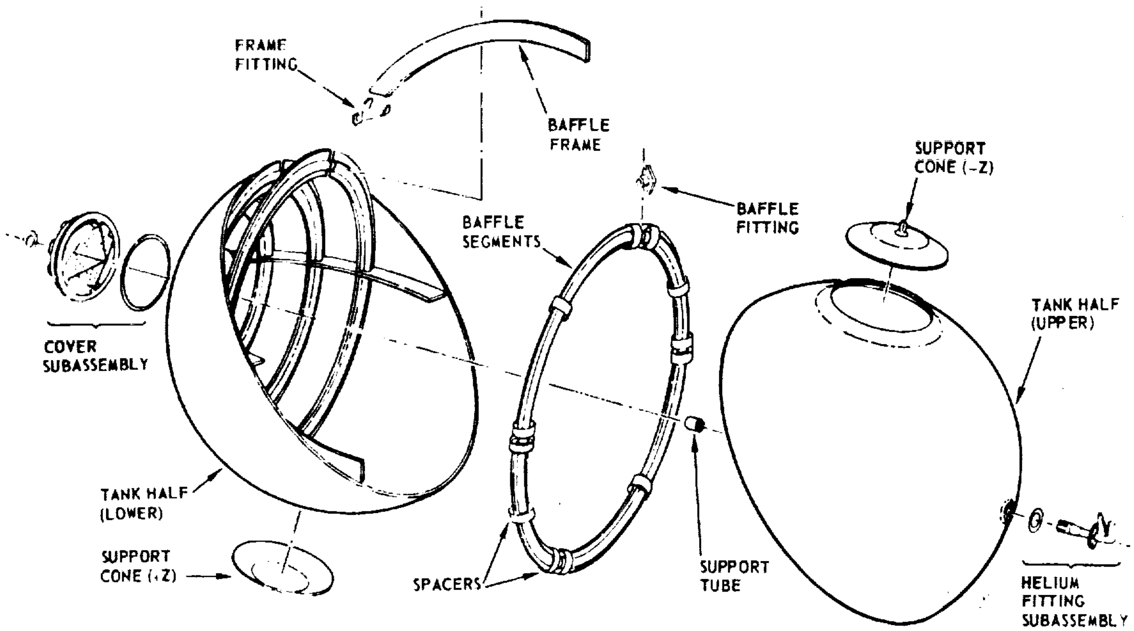


Figure 1 Propellant Tank Illustrating Typical Baffles, Entry Ports, and Attachment Fittings

a. Elastomeric and Metallic Bladders - Bladders can be fabricated from metallic and/or nonmetallic materials. They can be configured as a collapsing or expanding bladder.

Figure 2 presents a sketch of an expanding polymeric bladder design. The standpipe supports the expanding bladder and acts as a distributing diffuser for the pressurization gas. The propellant drain is situated at the base of the standpipe and the overflow drain port (if necessary) is located at the top. With a collapsing bladder, liquid expulsion is accomplished under the pressure of a gas introduced into the volume between the bladder and the tank wall. A collapsing bladder is currently preferred over an expanding bladder for the following reasons: higher expulsion efficiency, less probability of a flexural failure, and more operational experience.

b. Elastomeric and Metallic Diaphragms - Diaphragms normally accomplish expulsion through a complete reversal of the barrier geometry, with translation produced by gas pressure in the ullage volume. A polymeric diaphragm undergoes less severe wrinkling and folding during expulsion with an attendant increase in cycle life of two to five times that of an equivalent bladder.

Diaphragms are usually molded in a hemispherical shape and clamped equatorially around their perimeters by the two halves of a tank. Gas pressure on one side of the diaphragm serves to push the diaphragm in the opposite direction, expelling the fluid on the other side. A typical diaphragm system in a filled state is shown in Fig. 3.

c. Bellows - Bellows are corrugated cylinders which have a wall thickness that is relatively thin compared to the wall thickness of the tank in which they are used. The expulsion is accomplished by varying the internal tank volume by the accordion action of the bellows. Expulsion can be achieved by contracting the bellows with a higher pressure on the exterior as shown in Fig. 4.

d. Pistons - Pistons are used for expulsion devices where the pistons are forced to travel the tank length. Sealing is accomplished by rings on the piston which are in sliding contact with tank. The pistons can be held against the liquid by pneumatic pressure or mechanical actuation devices. Figure 5 shows a tank with a piston expulsion device.

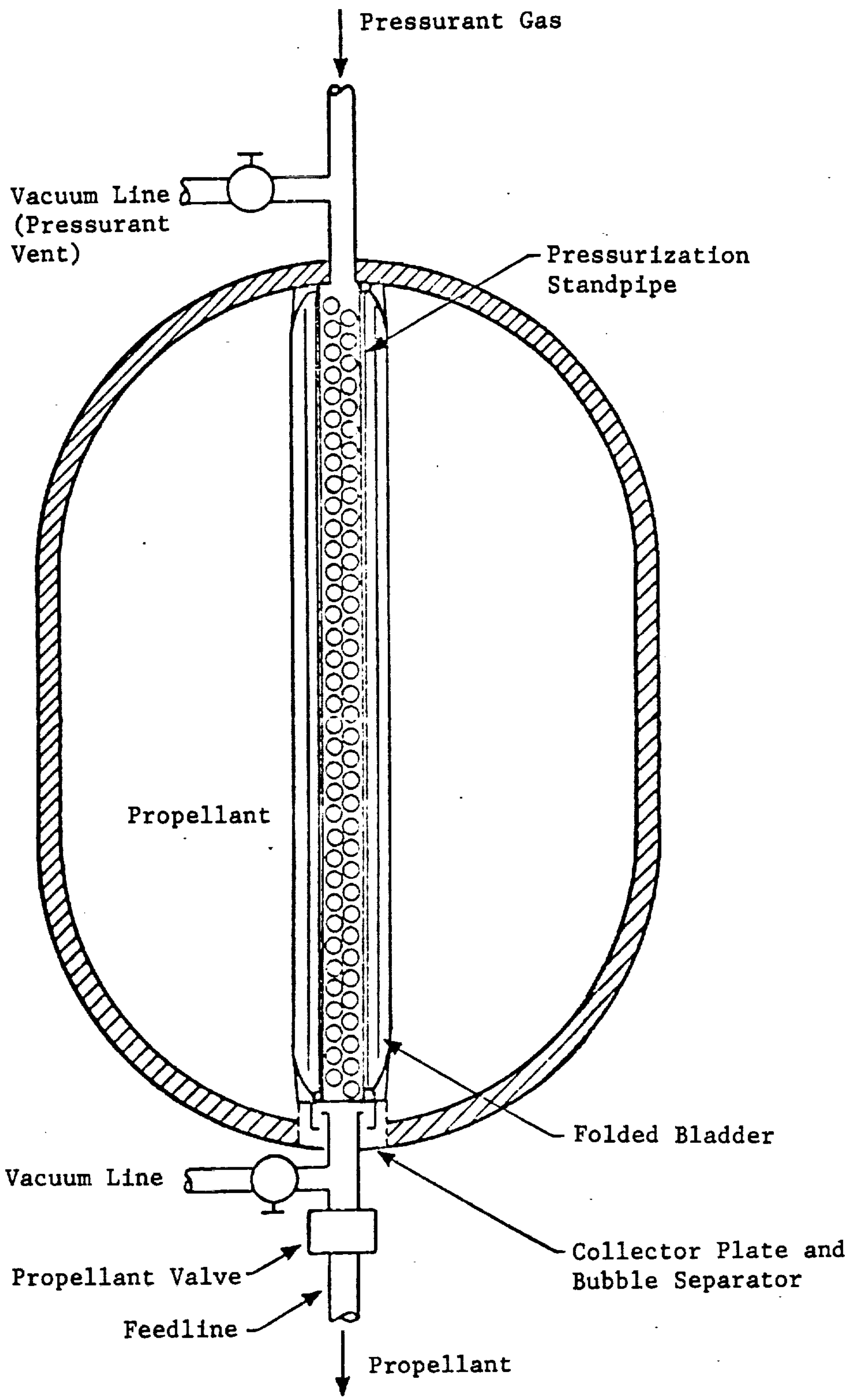


Fig. 2 Expanding Polymeric Bladder

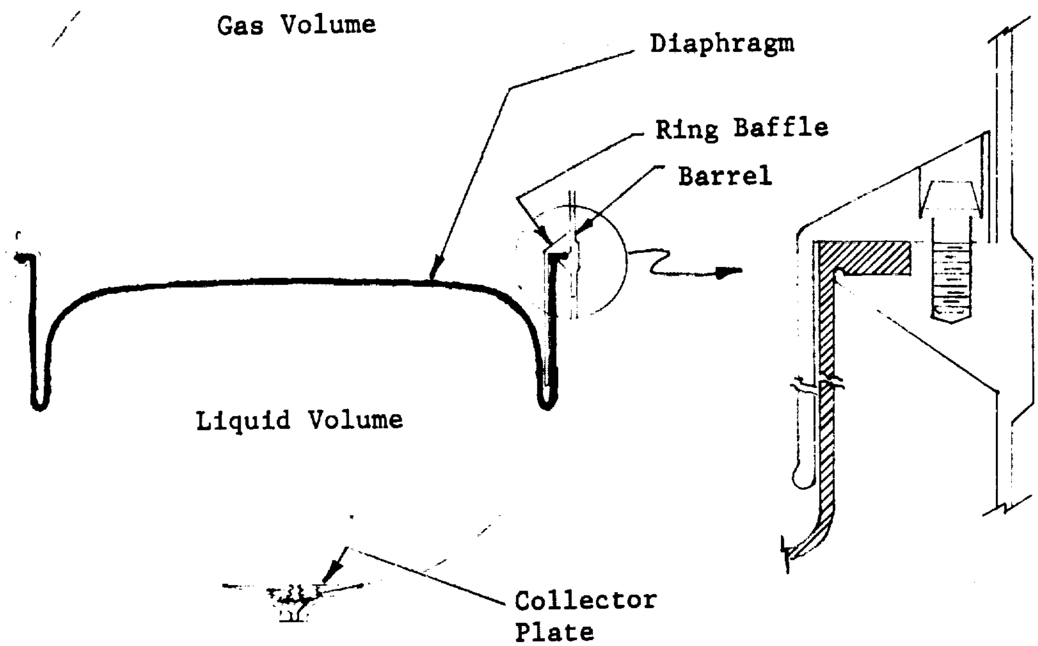


Fig. 3 Diaphragm-Type Tank

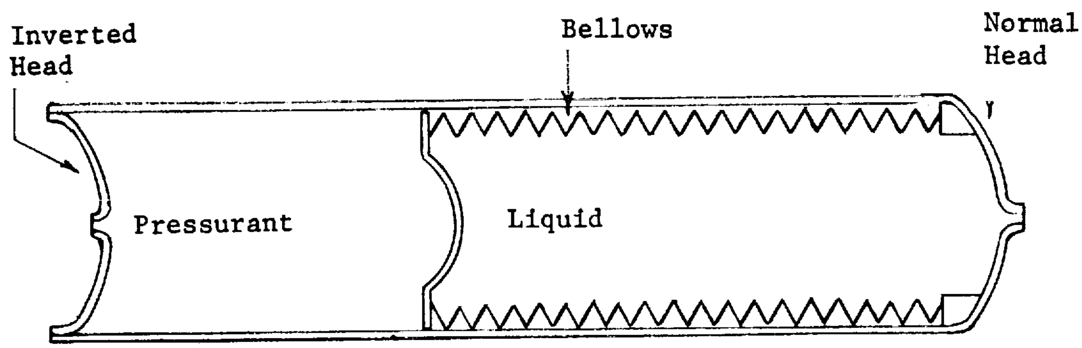


Fig. 4 Tank Using Bellow Expulsion Device

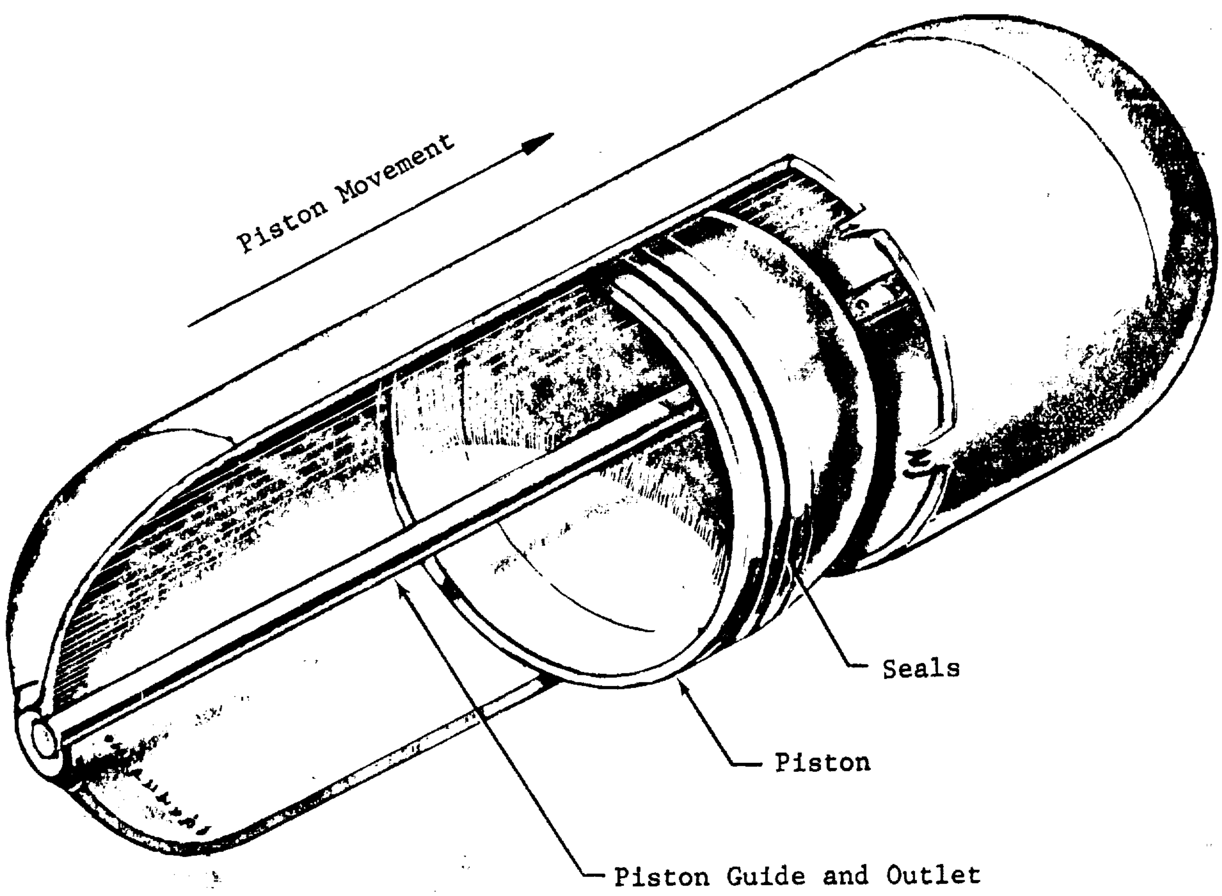


Fig. 5 Tank Using a Piston Expulsion Device

e. Surface Tension Devices - The screen is held tightly in place inside a tank by pleats, feet, or ribs, leaving a thin annulus all around the screen. The tank is filled as full as possible, making sure the screen is "wetted" and the annulus is filled. A pressurant gas is fed into the inside of the screen. Capillary forces set up a pressure difference across the screen that will push propellant through into the annulus but will keep gas inside. The various parameters, such as flow rate, screen pore size, etc, can be calculated from mission conditions. This system delivers gas-free propellant with no moving parts. Figure 6 is a detailed sketch.

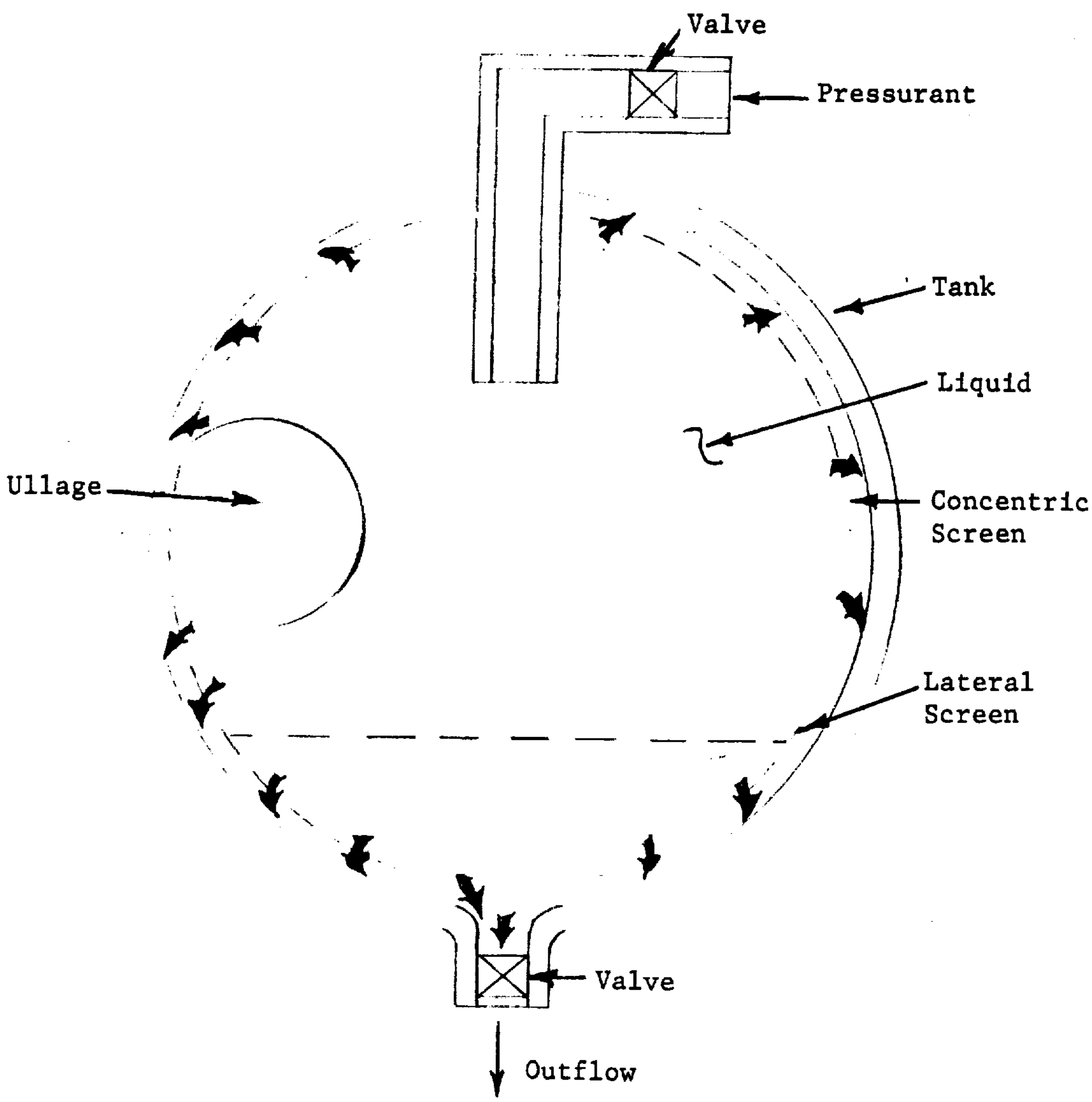


Fig. 6 Concentric Screen Concept

B. GUIDELINES FOR LONG-LIFE ASSURANCE

There are tank systems in existence that will perform successfully on a mission of up to 10 years duration. Care must be taken to fit the type of expulsion device best suited to the mission under consideration if a 10-year life is to be obtained. Tank failure modes which would prevent obtaining a 10-year service life are listed below in the order of their most probable occurrence:

1) Expulsion device leakage;

2) Leakage of the tank weld or leakage at fitting;

3) Tank rupture.

The estimated cycle lives of subject expulsion devices are as follows:

1) Convoluted, metal diaphragms are impermeable to propellant and work well for one expulsion. Severe metal working causes pinholing and failure, in most cases, when recycling is attempted.

2) Equatorially clamped, elastomeric diaphragms are good for 500 expulsion cycles with 99% expulsion efficiency.

3) Elastomeric and polymeric bladders, due to severe folding encountered during expulsion, have a cycle life limited to 30 to 35 cycles--shorter than elastomeric diaphragms lives, but longer than metal diaphragms lives.

4) Both metallic bellows and pistons are impermeable to propellants and both should function for more than 500 cycles.

5) Surface tension devices promise a life of at least 1000 cycles, many times longer than the other devices because they are immune to many of the failures that plague the other expulsion devices.

6) When exposed to corrosive fluids, the service lives of nonmetallic expulsion devices are less than three years due to permeation and incompatibility. The service life of metallic expulsion devices is greater than 10 years.

Table 1 summarizes the relative merits and weaknesses of positive expulsion devices for use in long-life space missions.

1. Design Guidelines

1) For a 10-year mission, no design changes are necessary in any of the designs studied to compensate for radiation damage. Resistance of all the materials to radiation is high enough and expected radiation levels are low enough to eliminate radiation damage as a problem.

2) Metal tanks, which represent proven designs and materials, shall be used for long-life applications.

3) Fracture mechanics tests are required to describe the fracture of materials under static, cyclic, and prolonged stress loading conditions.

4) Spherical tanks are required--constraints permitting. Spherical tanks offer the following advantages; they are: lightest in weight, easiest to fabricate, and more reliable for long-life due to less welds.

5) Use metallic expulsion devices, mission constraints permitting. They remain relatively unaffected when exposed to corrosive fluids.

6) Surface tension and metallic bellow devices have demonstrated characteristics which are consistent with long-life space applications. Surface tension devices are highly rated because they have no moving parts. Bellows are rated good because of past reliable performances.

7) The various methods of attaching the expulsion device to the inner tank wall do not present a problem for long space missions.

8) Use folding controls, such as masts or spring attachments on expulsion devices that exhibit potential problems with folding, ripping, and tearing failures.

2. Process Control Guidlines

1) Provide documented controls to assure handling and shipping care. Thin walled screened devices, metallic diaphragms, and bellows are structurally delicate.

2) Contamination controls for screened devices are necessary to obtain uniform operational characteristics from tank to tank. Require that vendors submit contamination controls and methods of implementation for screened devices.

3) Several tank users recommended that tanks be stored with an inert blanket pressure in an inert atmosphere for contamination considerations and the prevention of material corrosion.

3. Test Guidelines

1) To verify operational usage, tank selection criteria should advocate expulsion devices (elastomeric diaphragms and metallic bellows) for which positive tests can be performed.

2) Use radiography and dye penetrant test methods to verify the integrity of tank welds.

3) Conduct various combinations of propellant/elastomer exposure tests to develop long term compatibility. Ten year test data are not available.

Table 1 Relative Merits and Weaknesses of Expulsion Devices for Use in Long-Life Space Missions (Up to 10 Years)

PARAMETER	BLADDERS	ELASTOMER DIAPHRAGMS	METAL DIAPHRAGMS
Cycle Life	Metallic 1 cycle elastomer < 35	50 to 500	1
Calendar Life (Exposed to propellant)	30 days to 3 years for elastomer; 10 to 15 years for metallic bladders	30 days to 3 years	10 to 15 years
Resistance to Radiation (See Table 3)	Good in absence of oxygen	Good, no problems	Good, no problems
Attachment Problems	Strain on neck during slosh at neck flange	Equatorial clamping device very stable	Equatorial weld and clamping device very stable
Movement Failures (Folding, Tearing, etc.)	Severe folding causes tearing, pinholing, etc.	Slosh and expulsion may cause slight wear	Pinholing and tearing certain after first cycle
Permeability to Propellants	Severe problems with many propellants, especially N_2O_4	Problems with several propellants, again N_2O_4	Impermeable during first expulsion cycle
Compatibility with Propellants	Swelling followed by permeation	With new elastomers (Martin Marietta EPR 132), very little swelling (under 3%), some propellant breakdown	No problems when correct metal is used
Suitability for a 10-year Space Mission	Metallic bladders are unsuitable due to their limited cycle life and elastomer bladders are not suitable due to their life when exposed to propellants.	Elastomer diaphragms are not suitable for long life due to their limited life when exposed to propellants. The problem with elastomers is not necessarily that the elastomer degrades (EPT-10, for example, looks promising after several months exposure to N_2H_4) but that 10 year test data are just not available.	Unsuitable for more than 1 cycle

Table 1 (cont)

PARAMETER	PISTONS	METALLIC BELLOWS	SURFACE TENSION DEVICES
Cycle Life	>500	>500	>1000
Calendar Life (Exposed to propellant)	To 3 years due to seals	10 to 15 years	10 to 15 years
Resistance to Radiation (See Table 3)	Good	Good	Good, no problems
Attachment Problems	Pistons are held against the fluid by gas pressure or mechanical actuation device	Current applications have been limited to approximately two feet in diameter	Tight fit inside tank, very stable
Movement Failures (Folding, Tearing, etc.)	Non-metallic seals could wear or shred - metal seals could cause high breakaway function and wear	Metal crack and pinhole leaks	Not applicable
Permeability to Propellants	Elastomer seals are permeable to propellants	Impermeable	Control flow rate, pressurant gas can dissolve or become saturated in propellants.
Compatibility with Propellants	Teflon seals will swell when exposed to N_2O_4	No problems	No problem if correct metal can be fabricated into a screen
Suitability for a 10-year Space Mission	Metallic seals are suitable for a 10 year life; Elastomer seals are not suitable due to their life when exposed to propellants	Suitable for a 10 year life	Suitable for a 10 year life

C. LIFE LIMITING PROBLEMS AND SOLUTIONS

1. Failure Mechanism Analysis

Table 2 presents the Failure Mechanisms Analysis (FMA) which lists the failure modes for tanks and expulsion devices. The failure modes listed in order of their most probable occurrence are: (1) expulsion device leakage, (2) weld or fitting leakage, and (3) tank burst. Most tank failures are due to expulsion problems such as tearing, ripping or folding; permeation; incompatibility; expulsion device attachment; and radiation damage. Detailed information for these problems that result in expulsion device leakage is given in Subsections a through f. The tank leakage and burst failure modes are covered in Subsection f; some detail is given on the Apollo 13 tank failure. The long-life problems that result in the three failure modes and their potential solutions are discussed in the following subsections.

a. *Tearing, Ripping, or Folding Problems* - This type of failure indicates movement of some sort. This discussion applied primarily to bladders and diaphragms. Every failure from pinholing to actual tearing of a device that causes leakage on a scale larger than simple permeation will be considered.

Stretching and bag folding are the primary factors that cause breaks in balloon-type bladders. Stretching can be avoided by fabricating the bladder to be exactly the same volume as the inside of the tank. Folding, however, is a much more complex matter. In real bladders of even simple shapes there is a definite pattern to the folding as the bladder is collapsed. Severe stress areas are formed due to double and three-cornered folds and traveling creases (problem is described in detail in Ref 1). These stress areas are what eventually destroy bladder integrity.

Less elastic bladders (i.e., TFE-FEP Teflon) are pinholed or torn after a number of expulsions. After a longer time, more elastic elastomer bladders are similarly affected. Tests run using JPL's traveling crease machine showed that Teflon went through more cycles to failure as the thickness increased. TFE-FEP laminates performed the best of any materials tested with about 2000 or 3000 cycles to failure. Although good from a permeability standpoint, Teflon/aluminum laminates failed after less than 10 cycles. This shows that just one type of folding problem can cause failure.

Table 2 Failure Mechanism Analysis--Tanks, Pressure Vessels and Positive Expulsion Devices

Part and Function	Failure Mode	Rel. Rank	Failure Mechanism	Detection Method	How to Eliminate/ Minimize Failure Mode
Tank - positive expulsion device maintains medium over the tank outlet and provides medium upon demand.	Does not maintain medium over tank outlet due to:	1			
	Leakage of; a) diaphragm, b) bladder,		Both metallic and polymeric bladders and diaphragms experience material failures due to tearing, pinholing, wrinkling and folding caused by high stresses. (Metallic material problems also include rupture, material fracture, and minor cracks.)	Preinstallation Internal leakage can be measured by delta pressure across the expulsion device and monitoring pressure. Mass spectrometer tests.	Attachments between individual assemblies and the tank shell must provide sufficient structural compliance to avoid high stress interactions due to pressurization and cycling.
	c) bellows,		Bellows problems include leaf weld failure and weld pinhole leak.	Bubble testing is used to check on the capillary acceptability of both screen installation and relevant structure joints.	
	d) surface tension device,		Surface tension problems include the loss of trapped propellant and gas penetration of the metallic screen.		
	e) piston seals		Leakage past the piston or bellows seals due to wear or shredding of the elastomer seals.	These devices can be cycled both before and after installation.	Don't use dynamic seals for long-life applications.
	or expulsion cannot be performed because of;				

Table 2 (cont)

Part and Function	Failure Mode	Rel. Rank	Failure Mechanism	Detection Method	How to Eliminate/ Minimize Failure Mode
	a) cocking or siezure of the piston, b) buckling of the bellows or stuck in one position.		Metallic seals increase the breakaway frictional force.	Post-Installation Temperature and pressure measurements on the tank assembly.	
Tank - contain gases and fluids	Slow loss of medium due to weld leakage or fitting leakage.	2	Stress corrosion cracking, hydrogen stress cracking, voids or cracks in welds, incomplete seal of mechanical fitting.	Preinstallation Penetrant test x-ray test, neutron radiography, and mass spectrometer tests. Post-Installation Temperature and presure sensors on the tank assembly.	Control of weld characteristics and repeatability. Select a design which has a minimum number of welds.
	Rapid loss of medium due to tank rupture.	3	Overpressurization of tank.	Preinstallation Proof pressure tests. Post-Installation Temperature and pressure sensors.	Safety factors, choice of materials and pressurization controls.

Masts, spring attachments, more complex bladder attachments, and other type of folding control apparatus may help to lessen this problem. However, folding to some extent must be present in a bladder system.

Nonmetal diaphragms have two separate periods when there is danger to diaphragm integrity: (1) when the tank is filled and the diaphragm is raised to a point slightly above the equator of the tank, and (2) during and after expulsion. The problems involved are not as severe as the bladder problems simply because a diaphragm reverses its position instead of collapsing.

The first of the two periods has diaphragm wear as its main factor. In the raised or full position, the diaphragm is squeezed into a space that is too small. The excess of material rubs against itself because of cramping and sloshing, when the elastomer folds over on itself. Recent development such as the Teflon-coated baffle ring and proper ribbing of the diaphragm have reduced this problem to the extent that it does not now measurably cut diaphragm life or introduce contaminant particles from wear.

During expulsion, other wear and folding mechanisms become apparent. Traveling creases and many small wrinkles are formed by hydrostatic pressure (as is explained in Ref 1). The damage, however, is minimal, because elastomers resist permanent creasing. At the end of the expulsion cycle, the bladder can be cut through as it cuts off propellant flow by extruding into the outlet part. A sievelike barrier plate over the outlet port prevents this. To prevent the holes in the plate from cutting into the diaphragm, the diaphragm material should be thickened near the outlet part making the ratio of diaphragm thickness to hole diameter (t/d) greater than one, as explained in Ref 2.

Metal diaphragms can tip and buckle even during the first cycle, thereby causing leaks. Antitip devices are of some help, but better fabrication techniques are the real solution. In any case, the second expulsion cycle will cause this buckling. Thus, metallic diaphragm integrity can be maintained for only one cycle.

b. Permeation Problems - Permeation, in one form or another, is one of the main problems in positive expulsion systems today. The expected propellants to be used on the Shuttle Program include: liquid hydrogen (LH_2) and liquid oxygen (LO_2) for the main engine system, monomethylhydrazine (MMH) and nitrogen tetroxide (N_2O_4) for the orbital maneuvering system, and hydrazine (N_2H_4) for the

reaction control system. Materials selected must be impermeable (or tolerant) to the fluids involved, be able to be fabricated into an expulsion device, and function for a 10 year life. As in the case of Teflon metal laminates, this is difficult. If a metal diaphragm leaks at all, the metal structure has failed. Even in the case of screens, permeation must be kept at the desired level.

Permeation is a relative thing, depending on both the fluid and the barrier. Only metal barriers seem to contain the oxidizer, nitrogen tetroxide. Although all-metal diaphragms might work to expel this fluid, the weaknesses of the system discussed previously makes the search for something else necessary. Ethylene propylene and butyl rubbers are highly permeable to fuels such as unsymmetrical dimethylhydrazine (UDMH) and hydrazine for extended periods, and their compatibility is questionable. Nitrogen tetroxide permeates Teflon and other elastomers and polymers so much faster than do most fuels that, if an answer can be found for this oxidizer, it may be the general answer needed for most propellants now in use (see Ref 3 for more detail).

Nitrogen tetroxide is the big problem to be solved in relation to permeation. Teflon is virtually no barrier to it (permeation rates as high as 3 mg/in.^{2}/hr), and rubbers are little better. Chemically plated metal barriers may be removed by the oxidizer. Adhesives incorporated in a Teflon/metal laminate may be incompatible (for more detail see Ref 4). Filled Teflon membranes or Teflon/metal laminates cut permeation, but fabrication is very difficult. Filled Teflon materials and metal/Teflon laminates that resist delamination are promising, but even these do not eliminate all permeation.

Permeation can be designed for and tolerated as evidenced by the Teflon bladders flown in the N_2O_4-MMH tanks of Mars Mariner '71. Permeation may be undesirable in certain applications. Permeation is not a consideration regarding capillary screens. Screens are essentially infinitely permeable to all fluids. Therefore, a screen system must simply be designed to live with permeability effects.

In the case of metal screens, permeation rate is one consideration, because the device is designed to pass fluid through its pores. This flow rate depends on screen pore size, which depends on mission conditions (tank size, pressure differential, gravity, etc).

When these conditions are known, extensive calculations will give pore size and flow rate. If fabrication is carefully done, oversize pores will not be created and the calculation will be valid.

Although the screens will not pass gas bubbles, any pressurant gas dissolved in the propellant will be passed. This dissolved gas is present, because the pressurant gas contacts the propellant directly. When the propellant is transferred to the engines, any pressure drop may cause small gas bubbles to form. Careful pressure control in fuel lines could ease this problem if the dissolved gas is not a problem in other ways.

c. Incompatibility Problems - Incompatibility refers to the chemical and physical reactions that take place between propellant and expulsion devices that result in a usually undesirable change in one or both. Permeation is a type of incompatibility. A rise in pressure due to propellant or bladder decomposition is another incompatibility. Incompatibility can cause failure for a variety of reasons, as described below.

Elastomers swell excessively and give off gases when reacting to hydrazine-type fuels and oxidizers such as nitrogen tetroxide. Butyl and ethylene propylene (EPR) rubbers seem to have the best resistance to propellants of any of the elastomers. Most rubbers tested swelled excessively; however, the changes in EPR were lowest. EPR rubbers (i.e., Martin Marietta's EPR 132) swelled less than 3%. Other rubbers such as fluorocarbon and fluorosilicone elastomers swelled up to 400%. For more exact details see Ref 2. Bearing these facts in mind, if the newest types of EPR rubbers are not used, the diaphragms will degrade and swell prohibitively for any space mission longer than at most a month. A diaphragm or bladder whose equatorial seam is separated, or a bladder that is soft or brittle, or a bladder that swells, leaks, and releases gases is unsuitable because it causes system failure. All of these conditions result from bladder/propellant incompatibility. However, the use of new EPR rubbers minimizes these tendencies.

Metals vary in their reactions with propellants, but, in general, for almost every propellant used today, there is a metal that is compatible with it. Metals solve many compatibility problems simply by being more resistant to propellants than polymers. For example, N_2O_4 is corrosive to many metals. Nitric Oxide (NO) is an impurity in N_2O_4 that acts as an inhibitor in reactions between N_2O_4 and titanium. NASA specification MSC-PPD-2B, issued August 1, 1968, (supersedes MSC-PPD-2A) indicates that 0.8 ± 0.20

percent by weight of NO must be used with N_2O_4 in titanium tanks. Aerojet-General report LRP 198 (Ref 5) states that dry nitrogen tetroxide (less than 0.1% water) is noncorrosive to mild steel as well as aluminum, and that stainless steel alloys are not affected by N_2O_4 containing water at concentrations that are intolerable from a performance point of view. However, the corrosion rate of aluminum alloys exposed to N_2O_4 increases in relation to the water content percentage.

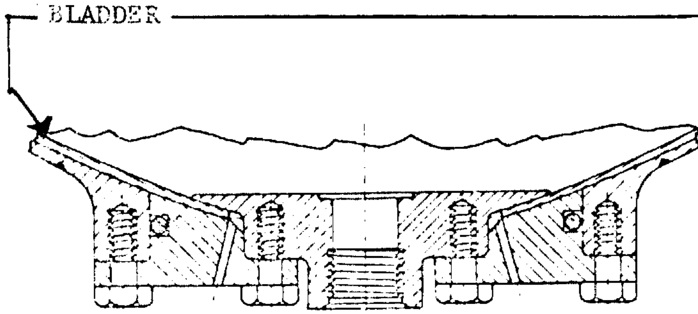
Although Teflon does not blister and generate gases in the presence of N_2O_4, it becomes very permeable to it. Subsequent swelling often breaks metal foil layers away from the bladder, thus defeating this attempt at impermeability. Hydrazine-type fuels react in a similar manner but to a much lesser extent. Thus, in the case of Teflon, incompatibility means permeation and severe swelling.

The survey of manufacturers and users did not reveal any particular compatibility problems with water, thermal control or hydraulic systems. Mr. D. Bailey of McDonnell Douglas indicated a handling problem of the bellows device which was used in the Skylab water tank system. The problem was solved by employing a bellows restraint to prevent bellows collapse during tank shipment.

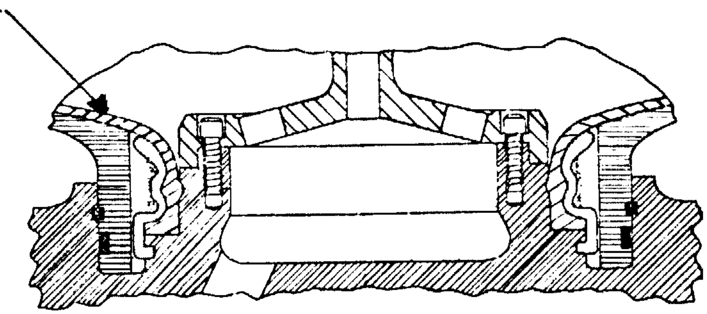
d. Expulsion Device Tank Attachment Problems - Although there are variations in the life capability of the various attachment schemes, this was one of the lesser problems under discussion.

Concentric, annular screens have no problems at all in this area. Fabricated with metal ribs or with pleats in a screen made slightly larger than the tank, the screen fits snugly in the tank without being attached to it. The gas inlet line is then the only real break in the screen, serving to orient the device inside the tank. Because screens do not move, there is no strain on any screen-to-tank attachment system.

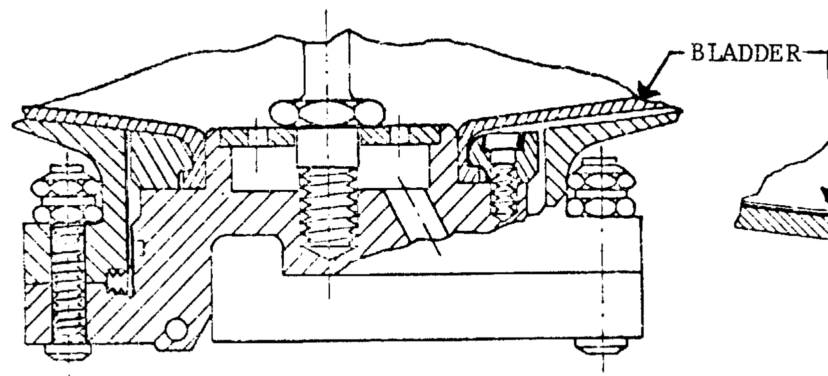
The attachment problems concerning bladders, although not major, are the most severe to be dealt with. The neck of the bladder must be strong, securely attached to the bladder and well sealed to the outlet connection by a separable tank flange as is shown in Fig. 7. During sloshing and expulsion, strain is concentrated on this small area, thus tearing is very possible if the bladder is not secure. To further reduce this strain, simple folding mechanisms as well as capacity filling should be used.



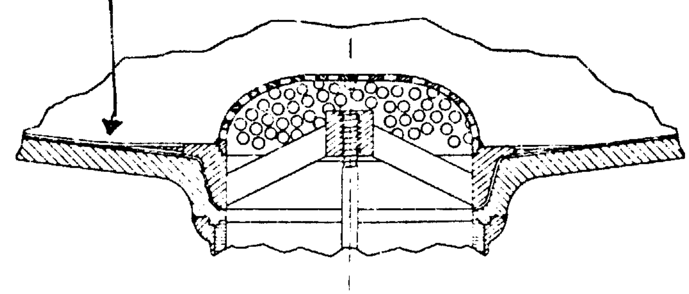
(a) *CORPORAL* DUAL GAS GENERATOR SYSTEM HYDRAZINE BLADDER



(b) *RANGER* BLOCK 2 MIDCOURSE PROPULSION SYSTEM



(c) *RANGER* BLOCK 3 AND *MARINER C* MIDCOURSE PROPULSION SYSTEM BLADDER



(d) *ALPS* FUEL AND OXIDIZER BLADDERS

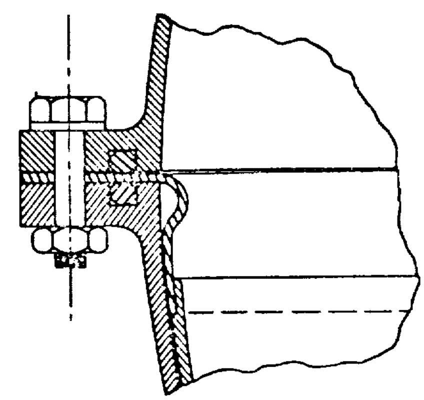
Figure 7 Bladder Neck Attachment Designs

Diaphragms are attached equatorially to a tank by clamping, bolting or welding the two tank halves around the perimeter of the diaphragm. (Metal diaphragms are often welded to the tank itself.) Figure 8 summarizes these methods. Nested metal diaphragms as in Fig. 8(c) have liquid on both sides and, thus, are very stable. If only one metal diaphragm is used, strain is greater and a stronger clamping and welding process is necessary. Elastomeric diaphragms are clamped between welded tank halves in much the same way. With the addition of an inner baffle ring to "collect" excess diaphragm material in the filled position, an attachment scheme much like the one in Fig. 8(d) has been used at Martin Marietta. The system resisted 400-psi differential pressure successfully under controlled temperatures and an inert gas welding process. Thus, the methods for securing diaphragms to their tanks appears to be quite dependable for a long space mission.

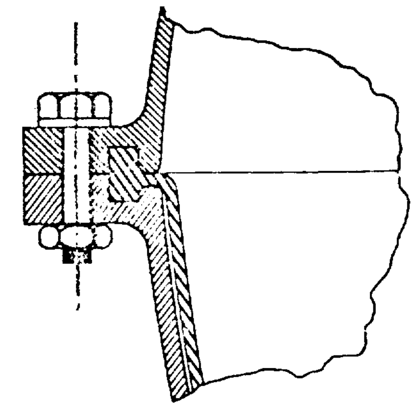
e. Radiation Damage Problems - As is shown in Table 3 and Fig. 9, threshold damage, or a 25% loss in properties, occurs at dosage levels higher than any to be encountered over a 10-year earth orbit mission for elastomers, polymers, and metals used in the fabrication of positive expulsion devices. In an oxygen-free system, with the shielding offered by fuel and tankage, the level is even less. Thus, radiation is not considered a problem on a long space mission for expulsion systems.

f. Weld, Fitting Leakage or Tank Rupture Failures - Weld or fitting leakage problems are more common types of failure modes than tank rupture failures when considering long-life. The Air Force Rocket Propulsion Laboratory at Edwards Air Force Base indicated that the main problems encountered during the Long-Term Storability of Propellant Tankage Tests were due to leakage in the weld area or the area adjacent to the welds.

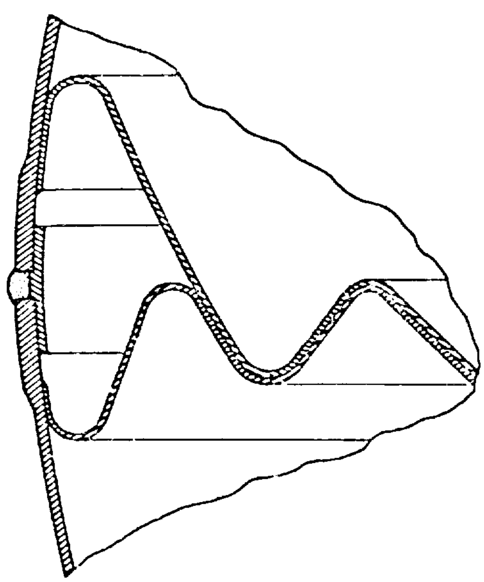
Environmental stress corrosion cracking is the phenomenon in which a crack propagates by stress corrosion cracking or hydrogen stress cracking. Stress corrosion occurs when a crack propagates by stress induced corrosion of the advancing crack tip in a corrosive environment while the metal is stressed in tension. The crack initiates at stress risers, for example, at the toes of butt welds, unfused roots of partial penetration welds, other geometrical discontinuities, and appear in the heat affected zone of the welds in most cases. Hydrogen stress cracking occurs when the hydrogen goes into a solid solution introduced by electrolytic charging, pickling, heat treatment, and by corrosion reactions.



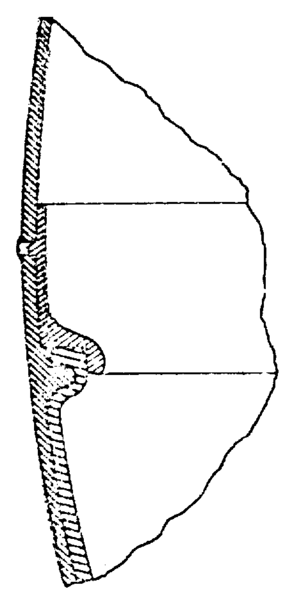
(a) METAL DIAPHRAGM OR RUBBER DIAPHRAGM BONDED TO METAL RING



(b) RUBBER DIAPHRAGM WITH BEAD CLAMPED BETWEEN TANK HALVES



(c) NESTED METAL CONVOLUTED DIAPHRAGMS WELDED INTO TANK



(d) RUBBER DIAPHRAGM WITH BEAD BONDED TO BACKUP RING AND CLAMPED BETWEEN WELDED TANK HALVES

Figure 8 Methods of Fastening and Sealing Diaphragms

*Table 3 Radiation Necessary to Cause Appreciable Loss of Properties in Polymers and Elastomers Used in Bladder and Diaphragm Fabrication**

Material	Dosage Necessary to Affect Properties (rad)	Suitability (Based on Radiation) for Long-Life Use in Bladders
Teflon TFE (Polytetrafluoroethylene)	4×10^4 in presence of O_2; 10^7 in absence of O_2	No problems for at least a 10-year mission; fillers improve resistance slightly
Teflon FEP (Fluorinated-ethylene-propylene)	10^6 to 10^7 (crosslinking at these dosages actually improves some properties)	No problems for at least a 10-year mission; better in this respect than TFE
Butyl Rubber	10^6 to 10^7	No problems for at least a 10-year mission
EPR Rubber (Ethylene-propylene)	10^6 (similar to butyl)	No problems for at least a 10-year mission
Most Other Elastomers (rubbers)	As much as 1 to 2 orders of magnitude greater than butyl rubber (10^8 or even 10^9)	No problems for over a 10-year mission (loss in properties almost negligible)

*See Ref 6 thru 8 for data given in this table.

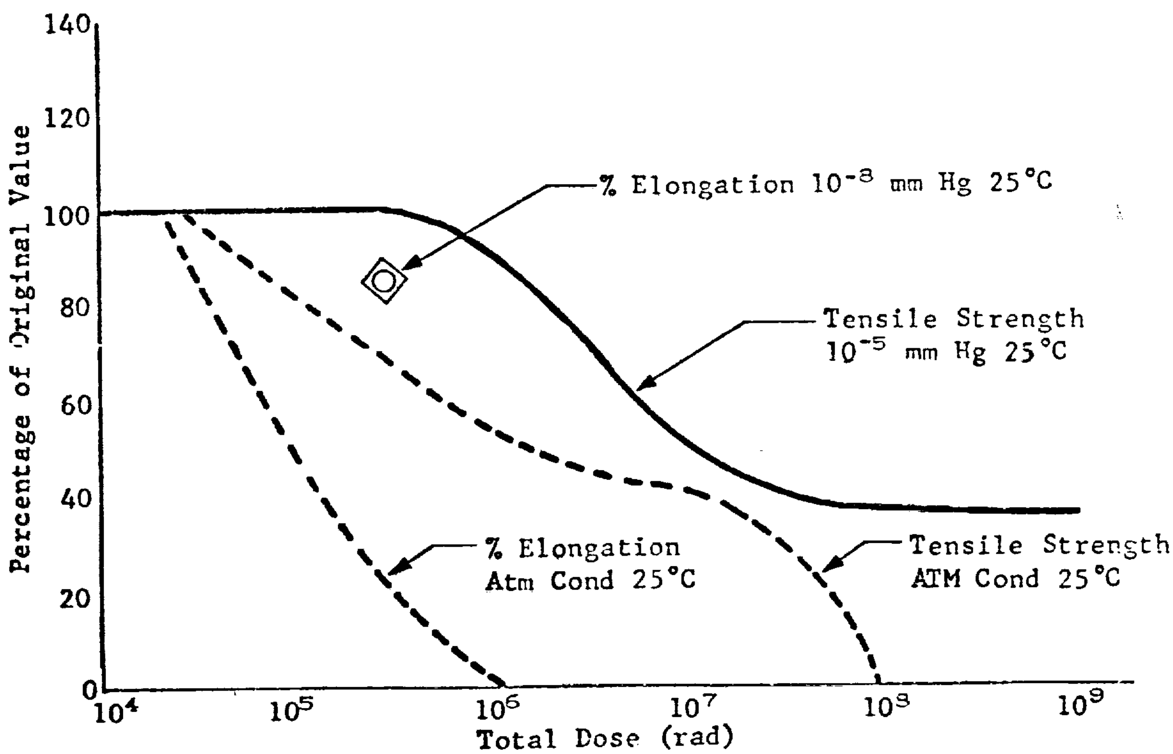


Figure 9 Tensile Strength and Percentage of Elongation, TFE Resins (Ref 6)

Hydrogen causes delayed failure under static loads with high strength materials. The embrittlement affect increases with deepness of the flow or notch. Solutions to these types of problems in tanks involves designs that would eliminate or minimize the factors that promote environmental stress corrosion cracking. Designs should avoid crevices, deep recesses, sharp corners, notches of any kind, and dissimilar metals.

Tank rupture failure modes are unlikely, in that they may be detected during normal tank testing. Some details from Ref 9 of the Apollo 13 tank failure are listed in the following paragraphs. The O_2 tank was manufactured by Airite Products and supplied to Beech Aircraft under subcontract. Beech completed the tank assembly, installing the quantity probe, motors, fans, and conducted tank assembly tests (see Fig. 10). The tank assembly was then supplied to North American Rockwell.

The Apollo 13 O_2 tank failure was due to the failure of the thermostatic switches that permitted the temperature of the heater tube assembly to reach 537.8°C in certain spots during ground operations. This heating damaged the Teflon insulation on the fan motor lead wire. During the manned flight, the fan motor lead wire, moved by the fan stirring, short circuited and ignited the Teflon insulation by an electrical arc. Combustion and overpressurization, then in the tank, probably failed the electrical conduit and the tank itself.

Recommendations to eliminate this type of problem included: (1) remove all wiring from contact with O_2, and unsealed motors which could potentially short circuit and ignite adjacent materials, and (2) eliminate the use of Teflon and combustible materials from O_2 and potential ignition sources.

After the Apollo 13 failure, NASA-MSFC requested that any high energy source used in the Skylab Program, be considered a Single Failure Point (SFP) in the Mission Level Failure Modes and Effects Analysis.

g. *Additional Studies Required* - There is no evidence that surface tension properties break down with age; however, this possibility must be investigated. Decreasing surface tension with time has been reported for a class of liquids. A comparable change in propellants could, in most cases, be compensated by overdesign.

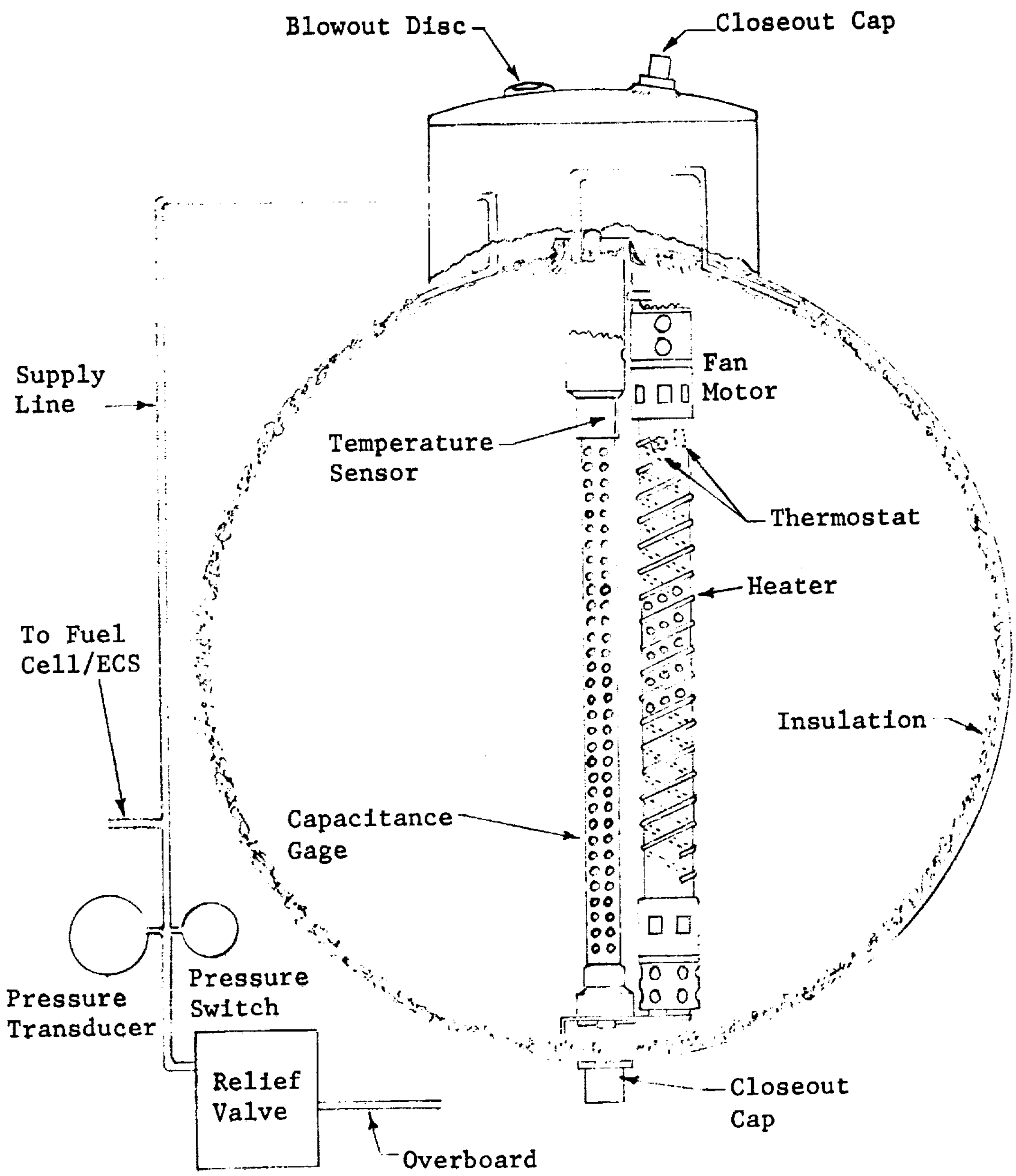


Figure 10 Apollo Oxygen Tank

2. Design

a. Selection Criteria - Considered as part of the design and selection criteria are the stress analysis, facture mechanics, design factors, and hardware life. A subsection concerning each topic as pertaining to long-life follows.

Stress Analysis - The mechanical properties of the material selected should be well established. The basic properties are located in MIL-HDBK-5 and the Aerospace Structural Metals Handbook. It has been customary to compile material strengths, either as single, nominal values (the practice of most material suppliers) or as single recommended minimum values as presented in MIL-5 Handbook.

A rigorous stress analysis is necessary for each tank design to establish the complete structural integrity of the tank for operational and long-life applications. This analysis should include all the critical loadings imposed on the tank during handling, testing, and operation. Tanks contain areas where changes in thickness and/or curvature occur, which produce additional stresses on the membrane. These stresses result from the additional forces (shear and moment) required in these areas to resist deflection and rotation (Ref 10).

Another condition to be investigated in detail is the local stresses imposed by concentrated loads. Concentrated loads in areas used for attachment devices can be expected.

Statistical methods are used to analyze the variability of imposed loads and material strength. The statistical data can be displayed as shown in Fig. 11. It is evident that the statistical approach permits calculation of the probability that the tank will successfully operate during a mission and quantitatively predict the load that will cause failure.

Fracture Mechanics - A design practice used by most companies, and recommended by this report, involves the principles of fracture mechanics. Fracture mechanics provides a basic framework for describing the fracture of materials under static, cyclic, and prolonged stress loading conditions. Coupon testing in the various environments has established material flaw sizes below which flaw growth will not occur during operational use. Trouble free tank use dictates that flaw growth under sustained stress be eliminated. The permissible sustained stress level during the

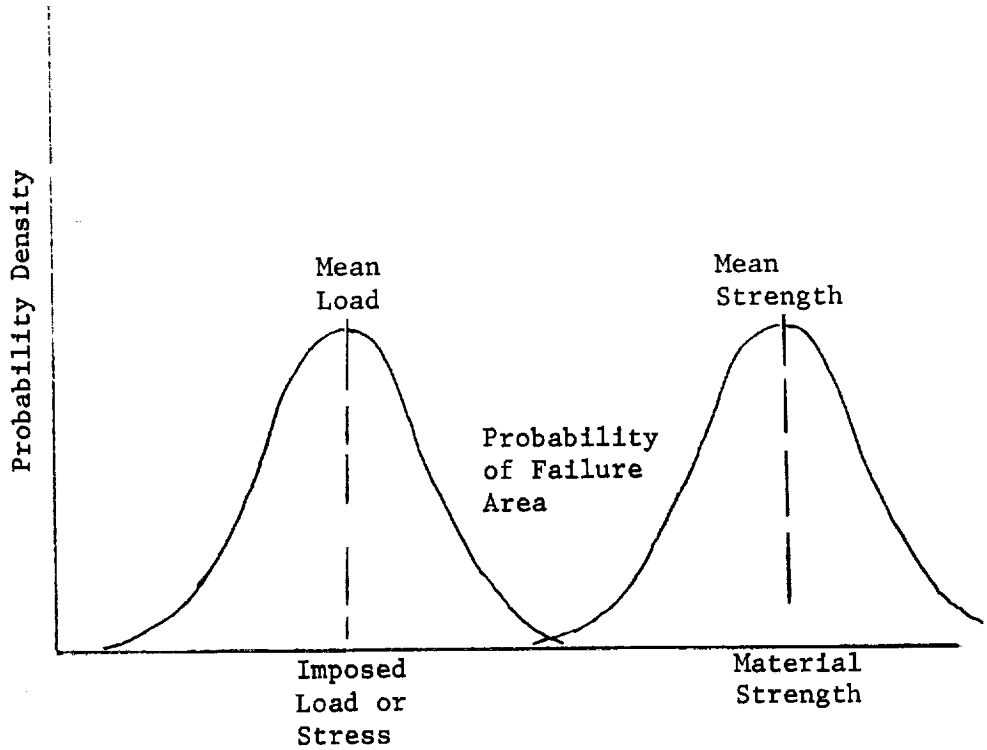


Figure 11 Stress/Strength Diagram (Warner Diagram)

service life is determined by the maximum flaw size which in turn is determined by the proof test stress level. A higher proof test stress will screen out smaller flaws and thus permits higher sustained stress levels in the various corrosive environments. The required sustained stress levels in the different environment, therefore, determine the minimum proof tests stress level. To this must be added an increment to account for flaw growth due to pressure cycling that can be expected during the life of the tank.

From the survey of the manufacturers and users, AiResearch indicated that they used fracture mechanic methods to develop the Astronaut Life Support tank for the Apollo program. This involved putting a defect (notch) in the heat effected zone adjacent to the weld. The tank was then subjected to pressure cycling until failure to prove that the failure was in a leak or rupture mode rather than a burst mode.

Other Design Factors - All metal tanks should be used for long-life applications since there are no particular advantages gained by the use of composite tanks. Composite tanks such as filament wound tanks weigh less than metal tanks, but when the attachments and associated hardware are considered, the weight savings would be small. Metal tanks are preferred due to:

1) higher reliability considerations such as leakage;

2) construction schedule would be longer for composites due to manufacturing techniques;

3) uncertainties that exist in composite construction stress calculations, and;

4) thermal expansion problems of composite materials.

Spherical tanks provide advantages over cylindrical and specially-shaped tanks. The advantages gained by the selection of spherical tanks include:

1) the sphere is the lightest of the geometrical shapes;

2) fabrication is easier and;

3) more reliable for long-life missions due to less welds.

In addition to Table 1, Table 4 lists some design factors for expulsion devices and tanks that are pertinent to long life. The long life advantages favor the all metal tanks because of material compatibility, loss of seal leakage, and the absence of permeation.

b. Results of Survey - The results of the general manufacturer and user survey for propellant tanks and pressure vessels are shown in Table 5. The results of the specific survey of tank users are shown in Table 6. Comments from the surveys are integrated into the text under the various sections.

Martin Marietta Expulsion Device Experience - This includes Titan, Viking, and various study programs. The Viking designs employ surface tension devices, the Transtage attitude control system used a bladder system, and the Transtage propulsion system used a retention method as outlined below.

The Transtage main propulsion system traps (Fig. 12) retain a propellant supply to ensure satisfactory engine start and operation after a coast period until the propellant is repositioned. The attitude control system is fired prior to start-up of the main Transtage engine restart resulting in propellant settling. Although not specifically designed for long-life, Transtage has performed multiple low-g main engine restarts, thus verifying the tank and retention device capability.

c. Alternative Approaches - The trade-off parameters include the selection of a moving expulsion device versus a stationary surface tension device or a metallic tank versus a composite tank. Material compatibility must be considered along with mission duration and requirements when developing the tank configuration. Table 4 presents some of the advantages and disadvantages to be considered in these trade-offs.

d. Apollo Helium Tank History - Two spherical tanks were used for Apollo Service Propulsion Subsystem (SPS) helium pressurization. The tank material chosen was titanium (6Aℓ-4V) based on the best strength-weight ratio. North American Rockwell designed the tank and Airite Corporation was selected to build and qualify the hardware.

On the Block I configuration, each SPS helium sphere was 40 in. outside diameter with a wall thickness of 0.450 in. Each tank had a volume of 19.4 cu ft and contained 48.5 lb of helium when pressurized to 4000 psi at 21.1°C. The tanks were designed to a maximum operating pressure of 4400 psi, 5850 psi proof, and 6600 psi burst.

Table 4 Expulsion Device/Tank Design Selection Criteria

EXPULSION SYSTEM	DESIGN FACTORS	
	ADVANTAGES	DISADVANTAGES
Piston	Positive displacement; variable initial ullage gas volumes; 98% efficiency of expulsion.	Heavy; leakage, jamming due to cocking or corrosion; limited to cylindrical tanks. Reduced reliability due to the possibility of shredding or wear of the sliding seals.
Elastomeric Diaphragm	Good expulsion efficiency (99 to 100%); trouble free and repeatable diaphragm geometry during expulsion.	Poor wear characteristics during prolonged propellant slosh; poor long life compatibility; limited to spherical tank geometry with significant initial ullage volume.
Metallic Diaphragm	Good compatibility; not sensitive to propellant slosh.	Limited to spherical tank geometry with significant initial ullage volume; high ΔP required for expulsion.
Rolling Metal Diaphragm	Good compatibility; not dependent on initial ullage volume.	High ΔP required for expulsion; limited to cylindrical tank geometry.
Bladder	Considerable development and flight experience; adaptable to most tank geometries and initial gas ullage volume.	Long term compatibility; gas permeation; poor expulsion efficiency (90%): folding geometry conducive to pinhole leaks; metal bladders are good for 1 cycle and elastomeric bladders are good for 100 cycles.
Metallic Bellows	Good compatibility; predictable performance adaptable to varying tank geometries and initial ullage volumes; expulsion efficiency of 98%.	High weight; cocking during expulsion; requires a larger tank to maintain propellant volume because of the volume it deplaces.
Surface Tension Devices	No moving parts; good expulsion efficiency (99 to 100%).	Failure may result from pore enlargement or an unexpected acceration forcing propellant from the tank trap.
Metallic Tank vs Composite Tank	There are no material compatibility problems with metal tanks, seal leakage is not a problem with metal tanks.	Composite tanks weight slightly less than all metal tanks.
Tank Geometry	Sphere is the lightest, spheres are more reliable because of fewer welds.	Not all expulsion devices can be used with one tank shape, available envelop may dictate tank shape.
Tank Testability	Subscale testing is possible on some expulsion device/tank designs.	Subscale tests are performed on materials, while the functional tests must be performed on the final unit.

Table 5 Results of General Manufacturing/Agency Survey - Propellant Tanks and Pressure Vessels

	QUESTIONS	CONSENSUS ANSWER
1.	Do you manufacture (use aerospace tanks? Usage?	Yes. Apollo, Skylab, Saturn, Minuteman, Atlas, Titan, Viking.
2.	What is the expected life of subject part?	Tanks should last for 10 years or more because metallic tanks are not life limited.
3.	What failure modes would prevent a ten year service life? What failures have occurred?	Leakage at welds, expulsion device cycle life, expulsion device material compatibility.
4.	What are the failure mechanisms (causes) of the failure modes?	Eliminate double-pass weld areas, welds which passed inspection but fail later and tight cracks.
5.	What solutions do you suggest for the above failure modes that would either enhance the operational life and/or increase the probability of success?	1) Use integral bosses 2) Avoid stress risers 3) Limit tank penetrations 4) Reduce the length and number of welds 5) Include in tank test programs, helium tests, X-ray examinations and use fracture mechanics tests.
6.	To achieve long life, what design features are incorporated in your tank?	Perform fracture mechanics tests and cycles test; this probably involves more tank weight.
7.	How do you determine part life?	If the tank assembly includes expulsion devices or motors, then these are considered the life limiting items. Most companies don't test for tank life but rely on field data.
8.	Did you test for specific failure modes and mechanisms and were any special testing techniques used?	Fracture mechanics techniques are used to assure tank will fail in a particular mode. Cryogenic proof tests and fracture mechanic concepts are used to detect the largest material flaw that the tank design can withstand.
9.	What process controls are necessary to ensure long life?	Weld repair techniques, handling methods, machine tolerances of hemisphere halves to permit quality weldments.

Table 6 Specific Survey of Tank Users

PART TYPE (CATEGORY)	USER	UNIQUE TEST OR SPECIFICATION REQUIREMENTS	RATIONALE/JUSTIFICATION
Propellant Tank Viking Program	Martin Marietta Aerospace	1. No tank qualification test. 2. Sizing test. 3. Fracture mechanic tests. 4. Liquid nitrogen proof test. 5. Pressure levels: 300 psi operative, 450 psi proof, 600 psi burst.	Unlike normal component development tests, the tanks are qualified at the subsystem level. Component confidence is gained through the sizing test where the tanks are pressurized near their yield strength and fracture mechanics tests employed.
Cryogenic Storage Tanks (Apollo Program)	Beech Aircraft Corporation	1. Require tank storage in an inert atmosphere. 2. Eliminate the motors from the oxygen tanks. 3. Performed burst tests at cryogenic temperatures (two) and at ambient temperatures. 4. Destruct forging tests. 5. Have used the oxygen tank without the fan since the Apollo 13 failure. They have determined that tank equilibrium can be maintained by heaters and temperature sensors. The vehicle movement normally supplies enough mixing until the 10% of the liquid remains, then hot spots may occur.	Inert atmosphere eliminates stress corrosion problems. An Apollo program requirement has been to eliminate Teflon and other combustible products from high pressure oxygen systems. Fracture mechanic tests should be performed on high pressure tanks to determine the actual limits that a particular material may withstand.
Coolant Tanks	AiResearch Corp. assembles the system.	1. Pressure tests a) Operating, 100 psig (max); b) Proof, 200 psig hydrostatic, at 21.1°C ±11.1°C on each part with the other part plugged; c) Burst, 300 psig. 2. External leakage, 1.10 x 10^{-5} lb/hr of MMS-602 at 100 psig (SCD and ATP require no visual leakage at 100 psig for 30 minutes.	Tank provides additional storage capacity for coolant to supplement the pump package reservoir and to provide the coolant system a higher tolerance against minor leaks. Tank has one inlet and one outlet fittings. Tank has metallic bellows assembly which acts as a spring piston.
Skylab Water Tank	McDonnell Douglas Corporation, Western Division	1. Required all metallic tanks with metallic bellows. 2. This tank was bolted together. 3. A design change involved a restraining hook to position bellows during shipment. 4. 200 bellows cycle test.	Metal construction eliminated age control products. The tank internal pressure is approximately 35 psig. The flight crew members have a sealing kit to patch leaks in case of tank failure.

Table 6 Specific Survey of Tank Users (concl)

Part Type (Category)	User	Unique Test or Specification Requirements	Rationale/Justification
Skylab Filament Wound Oxygen Storage Tank	McDonnell Douglas Corporation, Eastern Division	1. Pressure requirements: a) Operating 3000 psig; b) Proof 7530 psig at 70°F: c) Burst 10,000 psi. 2. External leakage, 2.1 x 10^{-3} SCC/sec GHe maximum at 4500 psig and ambient temperature. 3. Qualification cycling, 100 cycles, 100 psig to 4500 psig. 4. Reliability cycling, 1500 cycles, 100 psig to 4500 psig.	These tanks are cylindrical with elliptical heads and are made of fiberglass filament wound over a stainless steel liner. The liner serves as a pressure seal to preclude gas leakage while the fiberglass provides the structural integrity of the tank.
14.5 Inch Sphere used on the X-24A Program	Martin Marietta Aerospace	Specification test data included: 1. Die checks; 2. X-ray welds; 3. Proof pressure 5000 psi; 4. Design burst pressure 6600 psi; 5. Normal operating pressure 2850 psi; 6. Do not allow weld repairs after cryogenic stretch.	The long-term storage experience of Arde, Inc. Cryogorm 301 pressure vessels include: 1. Fabrication was completed on this pressure vessel November 29, 1966; 2. The vessel was exposed to extensive pressure cycling and environmental testing; 3. The pressure vessel is currently in use. 4. Tank repairs are made in the annealed condition (not after cryogenic stretch).

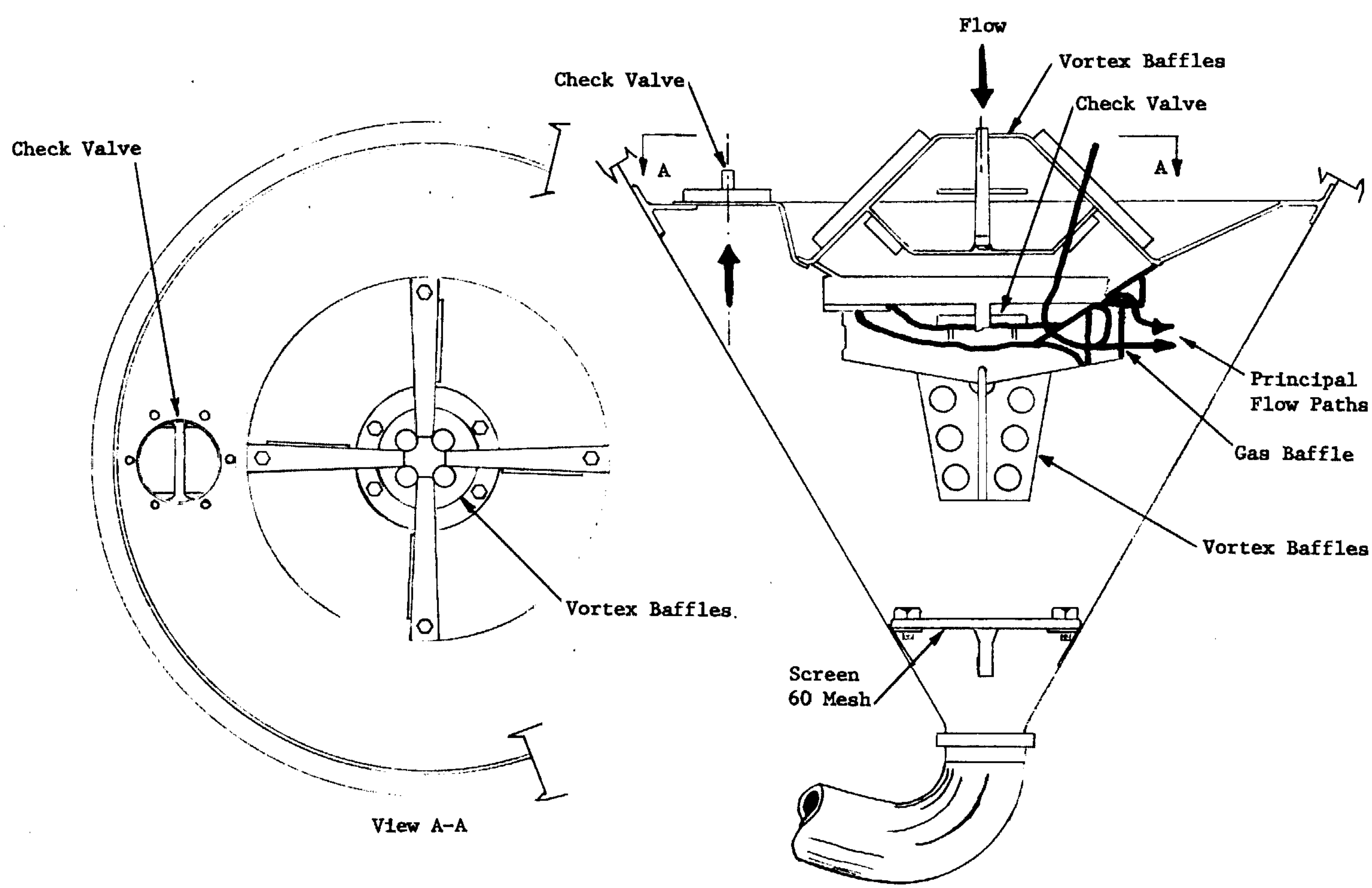


Figure 12 Transtage Propellant Tank

A major problem was encountered during qualification testing when two qual tank units burst prematurely below the required design burst pressure of 6600 psi. A design review was held at which time the decision was made to add two more qualification units to the program. These tanks were of identical design to the failed units except the exterior weld bead was removed flush to the outside diameter. Removal of the weld bead had the effect of allowing the weld joint to work more in unison with membrane during cycling. By alleviating the "belly band" fatigue stresses caused by the thicker weld section higher burst pressures were achieved as proven by test. The first SPS helium tanks were delivered and installed in the fall of 1964.

On Block II vehicles the SPS helium spheres were redesigned primarily to save weight. The tank material, outside diameter and methods of fabrication remained unchanged, however, the wall thickness was reduced to 0.366 in. Each Block II tank had a volume of 19.6 cu ft and contained 42 lb of helium when pressurized to 3600 psi at 21.1°C. The tanks were designed to a maximum operating pressure of 3685 psi, 4910 psi proof, and 5540 psi burst. Each tank weighed 316 lb, a savings of 77 lb per tank over Block I.

There were no problems encountered during qualification testing of Block II tanks which was completed in August 1966 (Ref 11).

e. *Hardware Life* - Each of the various types of expulsion devices has a life limiting characteristic when exposed to fluids for a specific time and a limiting number of cycles that can be performed before failure. Hardware life parameters are summarized in Table 1. These values are not exact, but can be used as a guide in the selection of a device for specific applications.

References 12 and 13 point out that verified mission lives of elastomeric expulsion devices when exposed to propellants is less than three years. The verified mission lives of metallic expulsion devices remain unaffected when exposed to propellants and are considered excellent for long-life applications. The following paragraphs detail the life cycle limitations of the expulsion devices. The cycle life data was extracted from Ref 12, 13, and 14.

Convoluted, metal diaphragms, fabricated in the folded position, are only good for about one cycle. The metal is worked so severely during this expulsion cycle that the surface may stretch and buckle.

Any attempt to recycle magnifies this metal working and, although the first expulsion cycle is usually successful, the second almost always causes leaks (see Fig. 13). One exception to a one cycle life was reported by Mr. D. Gleich of Arde Inc. (Ref 15). A six foot diameter ring reinforced metal diaphragm tank withstood multiple cycles. A small leak opened in the diaphragm shell during the first cycle, but the test was continued until the diaphragm was completely reversed. The leak was repaired and the diaphragm was reversed three more times without any further damage occurring.

Elastomeric diaphragms are considered to be highly recyclable when used correctly. When properly fabricated (with proper ribbing) elastomeric diaphragms will neither fold severely nor wear against themselves to any appreciable extent. Because they do not buckle as do their metal counterparts, these elastomeric diaphragms are good for up to five hundred expulsions, with efficiencies of more than 99%.

Due to several problems the cycle life of bladders is less than other elastomeric expulsion devices. The cycle life of elastomeric bladder is less than 35 cycles. Folding problems are very severe because the bladders are completely expanded and contracted during each expulsion cycle. The accompanying strain and wear weakens the bladders. If the bladder has a seam, the binding material there may be incompatible with the fluid. Finally a few problems are noticed at the bladder-to-tank attachment, though they are minimal.

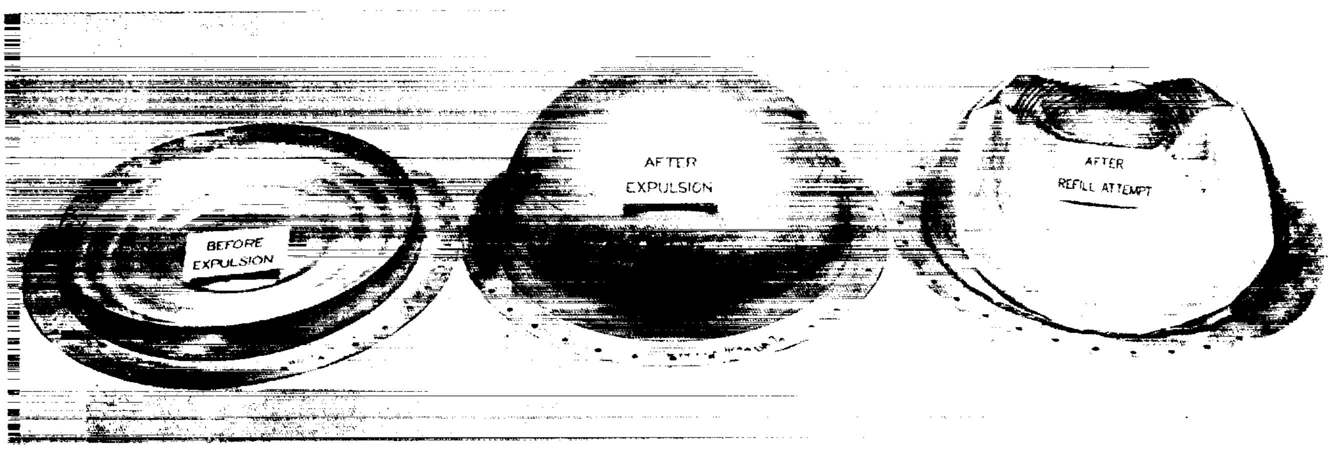
A piston type expulsion device is good for multiple cycles and can incorporate either metallic or nonmetallic seals. Both types of seals are subject to wear due to the sliding contact; however, the expected cycle life of piston expulsion devices is greater than 500 cycles.

Metallic bellows provide a nonpermeable membrane between the pressurant and propellant and can perform multiple cycle expulsions. Bellows are used in many applications and have the capability of functioning for more than 500 cycles.

There is a lack of data on cycles-to-failure for surface tension expulsion devices, however, lack of movement of the system eliminates many of the problems found in other devices. They even stand up well to vibration and slosh tests. Tests have shown that expulsion efficiencies of more than 99% for many cycles are possible. All indications are that the cycle life of surface tension devices is much higher than the other types.

f. Application Guidelines - Generally inflight maintenance has not been performed on high pressure vessels and propellant tanks. Mechanical fittings should be used at the tank interface so that a tank may be removed and replaced upon failure. An aspect that should be considered for long-term missions such as Shuttle, is resupply of expendables and methods to refill tanks. Bellows have no limitations in respect to refill, however, the refills of other expulsion devices are not desirable and requires vacuum loading techniques.

Interchangeability of tanks from one program to another has not been practiced, mainly due to the specific design requirements of each program.



(a) Before Expulsion (b) After Expulsion (c) After Refill Attempt

Figure 13 Metal Convoluted Diaphragms

D. TEST METHODOLOGY AND REQUIREMENTS

Proof testing of tanks is a standard method to detect material and manufacturing defects and to increase confidence that the tank is satisfactory. Common tank tests (some are applicable only to certain designs) include: External leakage, slosh, burst, expulsion device life, expulsion efficiency, and compatibility tests.

In reevaluating the material for the LM descent stage propellant tanks, Mr. R. A. Hilderman of Grumman Corporation (Ref 16) noted that, as its temperature is decreased, this titanium alloy shows a marked increase in unnotched tensile strength and a marked decrease in the notched tensile strength. It was concluded that a proof pressure test performed on the LM tanks at cryogenic temperatures would pass the tanks without flaws, and would fail the tanks with flaws. It is suggested that this test method be analyzed further to determine its effectiveness as a test method.

The environmental control system oxygen tanks manufactured by Brunswick for Skylab require a different test. These tanks are cylindrical with elliptical heads and are of composite construction with fiberglass filament wound over a stainless steel liner. The liner serves as a pressure seal to preclude leakage while the fiberglass provides the structural strength of the tank. The program qualification tests require 100 pressure cycles from 100 psig to 4500 psig. In addition, for reliability considerations of fatigue or laminant separation, 1500 pressure cycles from 100 to 4500 psig are performed.

Acceptance testing of bladders and diaphragms can be performed by leak checking, vibration testing, and expulsion testing. Since bladders and diaphragms are not passive expulsion systems, the polymeric material is degraded with each exercise, increasing the potential of failure on the next expulsion in spite of acceptable performance during the test. Consequently, vibration and expulsion testing for bladder or diaphragm acceptance are not recommended.

Leak checking bladders and diaphragms can be difficult because of polymeric material permeability and trapping of gas pockets in folds or between the device and tank hardware. If the expulsion system is leak-tested with a gas pressurant, it is hard to distinguish the bubbles expelled due to general permeation or from bubbles resulting from pinholes or tears.

An acceptance test based on a statistical approach decreases the confidence that a metallic diaphragm or bladder will perform satisfactorily. Only partial expulsion of about 20% of the loadable propellant volume can be demonstrated after the tank is assembled.

Both 1-g bench tests and drop tests have been used to verify the surface tension devices. Static and dynamic test apparatus (Ref 17) have been used to verify predicted performance. It is recommended that for long life applications, tanks selected with expulsion devices be limited to those for which positive tests can be performed.

The major limitation of long-term storage effects is the inability of conventional fluid/material compatibility criteria to predict leakage (Ref 18). Helium leak testing of tanks and the technique of leak testing are very important since small leakage which cannot be detected by X-ray or dye-penetrant inspection can lead to propellant leakage under adverse environmental conditions. These very small leaks can be detected through helium leak testing.

This study has generally noted that metallic expulsion devices are good for a 10 year exposure to propellants. The problems with elastomers for 10 year missions is not necessarily that the elastomer degrades, but that 10-year test data are just not available. It is recommended that various combinations of propellant/elastomer exposure be conducted to determine their long term compatibility.

The existing nondestructive test methods available include: radiography, die penetrant check, holography, portable isotope radiation sources, advanced two crystal ultrasonic methods, acoustic emission and eddy current. Neutron radiography is a method to locate hydrogen embrittlement in metals such as titanium, zirconium, and aluminum. In addition, this method shows weld areas which have the largest amount of hydrogen contamination. Martin Marietta uses radiography and dye penetrant tests to check welds because porosity and tight cracks can be more reliably tested. This report recommends the use of radiography and dye penetrant tests.

E. PROCESS CONTROL REQUIREMENTS

1. Screen Devices

Martin Marietta Advanced Manufacturing Technology group recently completed a program in which they developed and fabricated surface tension devices (Ref 19). The materials used were: (1) type 304L dutch twill stainless steel mesh count 250x1370/in.^{2}, (2) type 304L dutch twill stainless steel screen mesh 325x2300/in.^{2}, and (3) type 5056 dutch twill aluminum alloy screen 200x1400/in.^{2}. The program results are discussed in the next three paragraphs.

Joining of screen to screen and screen to nonscreen members was accomplished by employing several joining methods. All screen materials were easily resistance welded. Combinations of stainless steel screen to screen and screen to sheet can be satisfactorily fusion welded using a plasma needle arc welder. However, extreme care must be taken to ensure a good fit-up prior to welding. Aluminum screen can be joined by resistance welding and vacuum furnace brazing.

An important consideration is the influence of fabrication operations on the capillary retention of the screen device. Bubble point tests were performed on the screen material before and after thermal treatment and forming were accomplished. Thermal treatment did not impair the bubble point value, but forming a flat compound curvature core segment reduced the bubble point value. Although the thermal conditioning reduces the material strength, it does improve the handling and workability characteristics.

Pleating the tank retention screen assures structural strength, and no supporting members are needed. The expulsion efficiency is determined by the volume of the propellant trap. The pleated screen minimizes this volume while providing more surface area (than the flat screen) for propellant to enter the trap. The best rigidity is obtained with the pleats parallel to the warp wires (see Fig. 14).

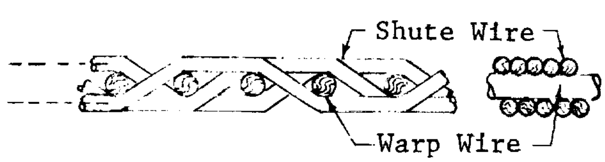


Figure 14 Dutch Twill Weave

2. Tanks

Reference 20 notes the problems associated with tank dome repeatability by an explosive forming processes. The deviations from planned contour were due to material effects of spring-back or bound-back. This required final sizing of the dome by elevated temperatures to achieve the desired dome contour. Though the thinout or thickness of the material may vary due to explosive forming, their tolerances are as good or better than other ambient temperature forming processes.

It is recommended that tanks be stored in an inert atmosphere and that an inert pressure blanket be maintained inside the tank to prevent contamination.

3. Expulsion Devices

It is recommended that handling care be instituted owing to the use of expulsion devices that are structurally delicate in the free-state. Applications of multilayered screens are subject to contamination. Bladders should be fabricated smaller than the tank to avoid tears and pinholes due to wrinkling. Shrinkage during curing must be controlled so that the bladder is large enough to seat against the tank wall without exceeding the material elastic limit.

Arde Incorporated has manufactured complete ring reinforced tank diaphragm assemblies using a 304L stainless steel diaphragm, with either 301 cryoformed or 304L tanks. Figures 15 and 16 show a ring reinforced diaphragm design. Aluminum diaphragms have been fabricated. However, there is a problem of joining the reinforcing rings to the diaphragm. Titanium cannot be used as a diaphragm material because of its poor elongation properties.

Bellows diameters have generally been limited to less than two feet. In addition to propellant compatibility, metal bellows must: (1) remain ductile after welding, (2) possess good formability, and (3) provide a high allowable strain to density ratio.

4. Welding

Martin Marietta study report (Ref 21) indicated that strict adherence to joint fit up and geometry are necessary when developing weldments free from cracks, mismatch, and warpage. Other conditions that are mandatory to welding practice and quality results to achieve long-life include:

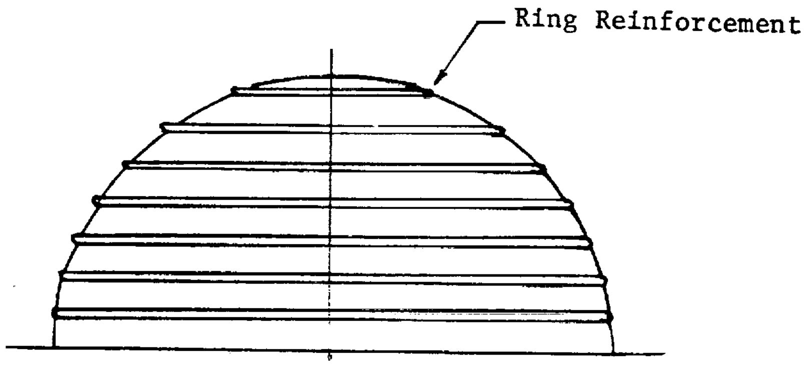


Figure 15 Metallic Positive Expulsion Diaphragm

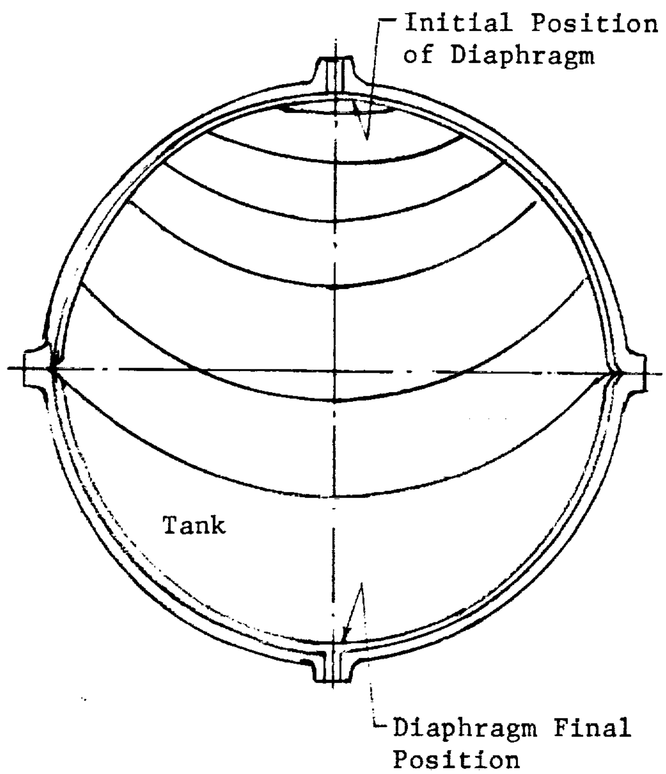


Figure 16 Diaphragm Reversal, Intermediate Positions of the Diaphragm are Shown

1) Gap free welds;

2) Similar thickness to both edges of the weld joint;

3) Cleanliness of joint area;

4) Minimum number of starts and stops;

5) Adequate weld tooling;

6) Automatic or semiautomatic welding process;

7) Interference fit for circular weld joints.

In the long term storability of propellant tankage test (Ref 16), it has been observed that double heat welds which occur at start/stop points and at weld intersections or at weld repairs lead to a high incidence of hot short cracks and leakage at these points. This condition is especially prevalent in manual weld repair due to poor control of heat input. It is concluded that quality control criteria for acceptance of welds, limit the number of weld repairs to four and minimize the number of double heat welds to be consistent with the geometrical configuration.

F. PARTS USAGE CONSTRAINTS

A review of Table 1, summarizing the strengths and weaknesses of each of the expulsion devices, shows that the surface tension devices and metallic bellows are superior to the other concepts in most ways. If only one cycle is considered, the metal diaphragm concept is only slightly less suitable on a long space mission. Factors such as incompatibility and permeation seem to rule out polymer bladders and some types of elastomer diaphragms for use on a long space mission.

G. REFERENCES AND BIBLIOGRAPHY

REFERENCES

1. R. N. Porter and H. B. Stanford: *Propellant Expulsion in Unmanned Spacecraft*. Technical Report 32-899. Jet Propulsion Laboratory, California Institute of Technology, Pasadena, California, July 1, 1966.

2. O. F. Keller and L. R. Toth: *Alps, Generant Tank and Cell Assembly*. Technical Report 32-865. Jet Propulsion Laboratory, February 28, 1966.

3. Stephen P. Vango: *Determination of Permeability of Cast Teflon Sheet to Nitrogen Tetroxide and Hydrazine*. Technical Memorandum 33-55. Jet Propulsion Laboratory, August 25, 1961.

4. Howard B. Stanford: *The Measured Permeability to Nitrogen Tetroxide of Some Potential Bladder Materials*. Technical Memorandum 32-123. Jet Propulsion Laboratory.

5. *Storable Liquid Propellants Nitrogen Tetroxide/Aerozine 50*. Report No. LRP 198, Aerojet-General Corporation, June 1961.

6. *Have G/Super Temp, All About Radiation and Teflon*, Technical Bulletin, Vol 63, Issue 9. American Super-Temperature Wires, Inc., December 1963.

7. Mary Jane Oestmann and J. R. Kircher: *Radiation Effects on Materials and Equipment*. 339B. Society of Automotive Engineers, 1961.

8. John B. Ribbenhouse and John B. Singletary: *Space Materials Handbook, Supplement 1 to the Second Edition, Space Materials Experience*. NASA SP-3025, ML-TDR-64-40, Supp 1. NASA and United States Air Force, 1966.

9. *Report of the Apollo 13 Review Board*. NASA, June 1970.

10. *Pressurization Systems Design Guide*. Report No. 2736. Prepared for NASA Contract NAS 7-169 by Aerojet-General Corporation, July 1966.

11. C. R. Gibson, *et al: History of the Apollo Service Propulsion System*. Propulsion and Power Division, Manned Spacecraft Center, Houston, Texas, January 1970.

12. *Investigation of Space Storable Propellant Acquisition Devices*. MCR-70-71, Vol I, Final Report, NAS 7-754, Martin Marietta, Denver Division, October 1970.

13. *Investigation of Storable Propellant Acquisition Devices*. MCR-70-171, Vol II, Final Report, NAS 7-754, Martin Marietta, Denver Division, December 1970.

14. *Handbook of Long-Life Space Vehicle Investigations*. M-70-16, Report No. T-70-48891-012, Martin Marietta, Denver Division, December 1970.

15. D. Gleich: *Six Foot Diameter Metallic Diaphragm for Fluid Storage and Positive Expulsion Systems*. Paper No. 70-683, AIAA 6th Propulsion Joint Specialists Conference, June 15-19, 1970.

16. R. A. Hilderman: *The Possibility of Cryogenic Proof Testing*. Paper No. 71-336, AIAA/ASME 12th Structures, Structural Dynamics and Materials Conference, April 19-21, 1971.

17. S. C. DeBrock, *et al: Survey of Current Developments in Surface Tension Devices for Propellant Acquisition*. Paper No. 70-685, June 15-19, 1970.

18. H. M. White, 1st Lt. USAF: *Long-Term Storability of Propellant Tankage*. AFRPL-TR-71-113, Edwards Air Force Base, Edwards, California, November 1971.

19. *Capillary Screen Technology*. Report No. T-71-48990-005, Martin Marietta, Denver Division, December 1971.

20. C. Caudill, R. Kirlin: *Improved Producibility of Titanium Tankage*. Report No. T71-44856-005, Martin Marietta Corporation, December 1971.

21. F. R. Clover: *Welding Techniques for New Alloy Pressure Vessels for Long-Term Storage of CLF_5*. AMT LAB Report 70-12-1, Martin Marietta Corporation, Denver, Colorado, December 1970.

BIBLIOGRAPHY

1. *Optran Program, An Analytical Study of Optical Propellant Transfer.* P-66-101 (Brochure 2). Martin Marietta Corporation, Denver, Colorado, November 14, 1966.

2. Donald J. Simkin (ed): *Chemical Engineering Techniques in Aerospace.* No. 62, Vol 60, American Institute of Chemical Engineers, 1964.

3. A. J. Bauman: *Some Design Considerations, Large Expulsion Bladders for Nitrogen Tetroxide and Hydrazine.* Technical Report 32-862. Jet Propulsion Laboratory, January 15, 1966.

4. *Aerovessels Product Areas.* Aerojet-General Corporation, Downey Plant.

5. T. R. Barksdale and D. L. Balzer: *Liquid Outflow Tests of Surface Tension Systems under Minus One-G.* TM-1661-66-12. Martin Marietta Corporation, Denver, Colorado, 1966.

6. *Joint AIAA/Aerospace Corporation Symposium "Low Gravity Propellant Orientation and Expulsion," Technical Session Proceedings.* May 21-23, 1968, Los Angeles, California.

7. *Welding Techniques for New Alloy Pressure Vessels for Long-Term Storage of CLF_5.* ATM Lab Report 70-12-1, Martin Marietta, Denver Division, December 1970.

8. C. Caudill, R. Kirlin: *Solid-State Bonded Liquid Rocket Propellant Containers.* Technical Report AFRPL-TR-71-69, Air Force Contract F04611-69-C-0056, Martin Marietta Corporation, July 1971.

9. R. E. Field: *Propulsion Requirements for Space Shuttle.* Paper No. 71-657, AIAA/SAE 7th Propulsion Joint Specialists, June 14-18, 1971.

10. C. L. Lofgren and D. Gieseking: *Cryogenic Positive Expulsion Diaphragms.* Boeing Company report for NASA Contract NAS 3-11204, Report No. NASA CR 72732, October 1970.

11. R. Botwin: *Flight Performance of the LM Ascent and Descent Propulsion Systems.* Paper No. 70-673, AIAA 6th Propulsion Joint Specialist Conference, June 15-19, 1970.

12. *Fracture Control of Metallic Pressure Vessels.* SP-8040, NASA, May 1970.

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XII. NICKEL-CADMIUM BATTERIES

by J. C. DuBuisson

XII. NICKEL-CADMIUM BATTERIES

A. INTRODUCTION

The three basic types of secondary batteries used in aerospace applications are: nickel-cadmium (Ni-Cd), silver zinc (Ag-Zn) and silver-cadmium (Ag-Cd). In general and from a long-life standpoint, the sealed Ni-Cd cells are the best of the three types mentioned. The reasons for this can be deduced from Table 1. This table illustrates the basic properties to be considered.

Although its energy density is low, (Figure 1) the Ni-Cd battery generally tolerates the other parameters of long space missions much better than the Ag-Cd and Ag-Zn cells. Ni-Cd batteries live longer because they are not nearly as chemically and physically self-destructive as the two cells involving silver. Disadvantages of Ni-Cd batteries include complex charge controls, and lower system electrical efficiency, poorer discharge voltage stability and charge retention. Only Ni-Cd batteries are considered in this chapter.

A typical sealed secondary cell is shown in Figure 2. The negative plate tabs are sometimes welded directly to the battery case, eliminating one terminal post and saving weight. However, the current trend is to use two terminal posts to reduce the direct effects of any shorts. Inert, highly porous sintered plates of nickel contain the active materials. In the charged state, the active material in the positive plate is nickelic hydroxide $Ni(OH)_3$, and the negative electrode contains metallic cadmium. The electrolyte is potassium hydroxide (KOH). The basic reaction involved in charging the battery is the oxidation of the positive plate active material to nickelic hydroxide and the reduction of the negative electrode to metallic cadmium.

Depending on battery temperature, state of charge and charging rate, the battery must be overcharged to overcome charging inefficiencies and restore full capacity. Oxygen is evolved at the positive plate during charge and overcharge. In sealed cells, provision is made for the oxygen gas to diffuse thru the separators to the negative plate where it recombines with the metallic cadmium. There is an excess of negative plate material to absorb

Table 1 Basic Battery Properties To Be Considered

Properties	Ni-Cd	Ag-Cd	Ag-Zn
1. Temperature Range (Optimum) (°C)	-20 to +30	-40 to +120	-40 to +165
2. Seals	Easily sealed	Sealing battery is questionable	Very difficult to seal
3. Cycle life	Very high	Medium	Low
4. High Shock Loads	Most likely to be harmed	Design changes could prevent breakage-potting, etc	Most resistant
5. High Deceleration Loads	Proper orientation of battery and proper separators prevent most problems	Same as Ni-Cd	Same as Ni-Cd
6. Speed of Dendrite Growth through Separator	Very slow	Moderately fast	Very fast
7. Gassing	Gas evolved, but most of oxygen recombines	Some gas evolved; venting usually used	Large amounts of gas evolved; venting very often used
8. Energy Density	Low	Medium	High

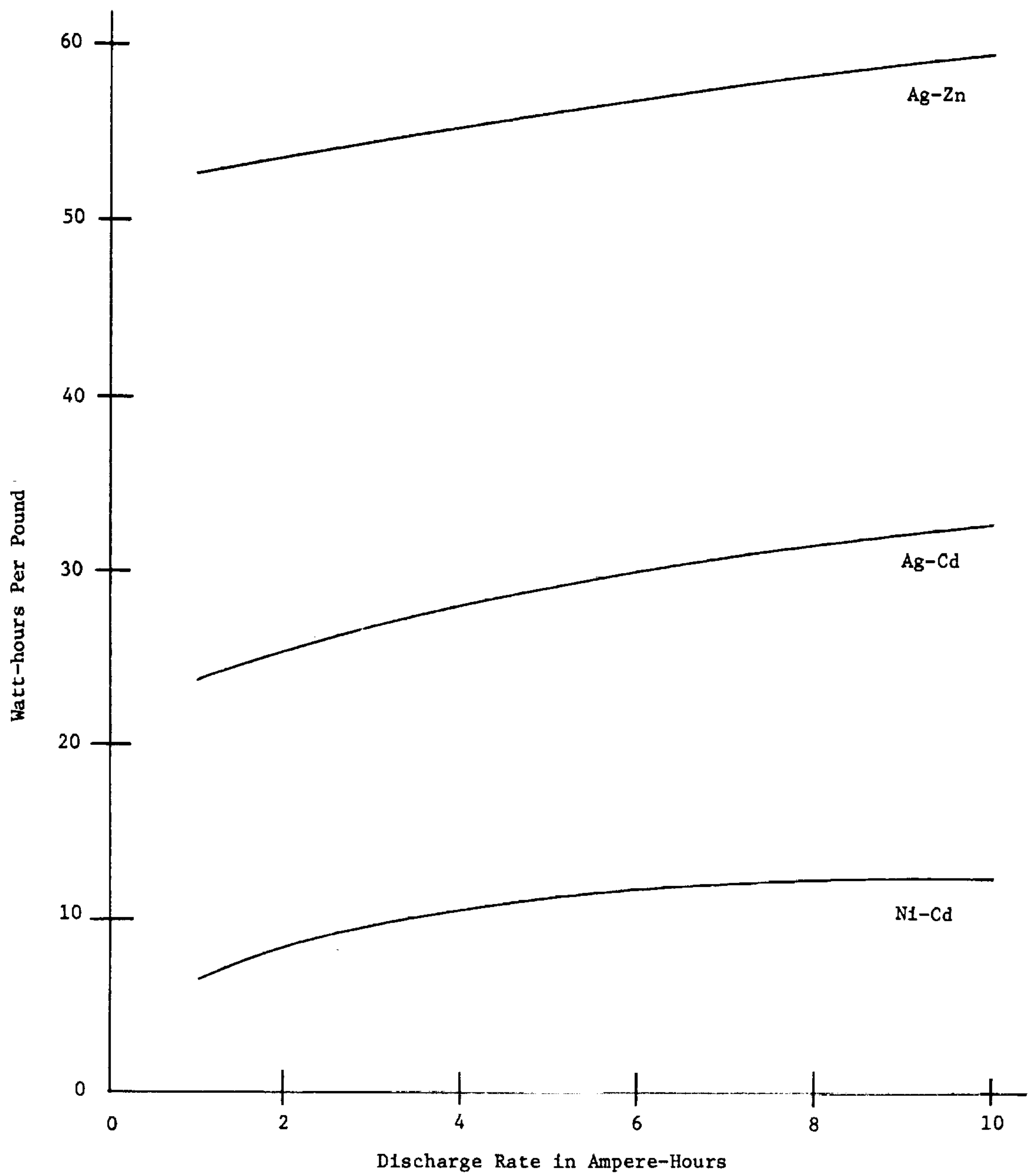

Figure 1 Typical Energy Density vs Capacity for Secondary Cells at Beginning of Life

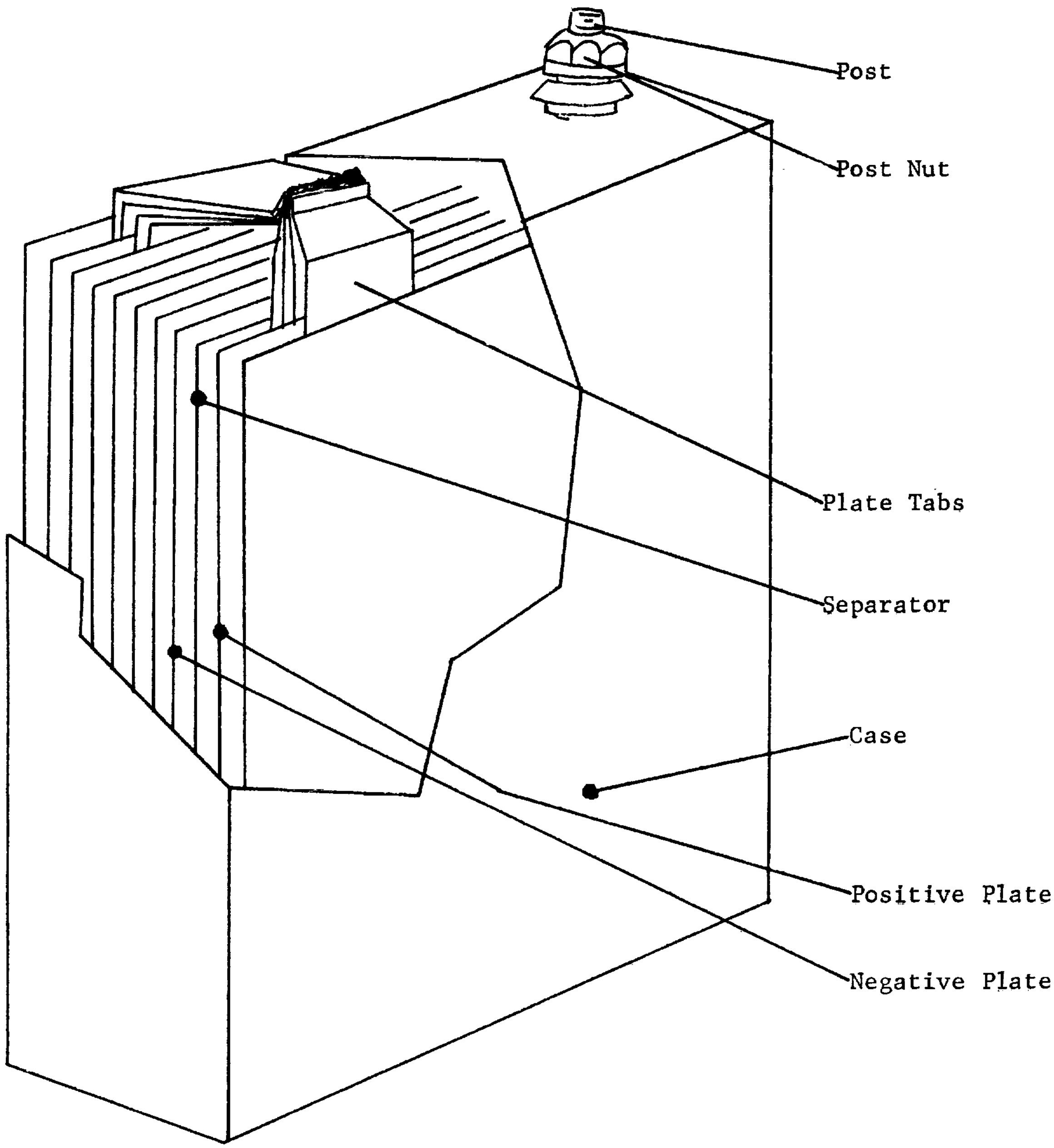

Figure 2 Ni-Cd Sealed Secondary Cell with Sintered Plate Construction

the oxygen and minimize the cell pressure increase during overcharging. Should any hydrogen evolve from the negative plate due to gross overcharging, a potentially dangerous (explosive) situation could develop. The KOH electrolyte is not involved in cell reaction, providing only ionic conduction.

Some characteristics of the Ni-Cd battery (as well as two other types of cells) are given below:

Battery Electrical Characteristics

Cell Type	Normal Operating Voltage (Plateau Voltage)	End of Charge Voltage (v)	End of Discharge Voltage (v)	Energy Density (wh/lb)
Ni-Cd	1.2	∿ 1.47 @ 21.1°C	1.0 (or slightly higher)	10 - 15
Ag-Cd	1.1	1.4	0.9 to 1.0	11 - 38
Ag-Zn	1.5	1.9	1.2	15 - 100

The reader is referred to the bibliography for sources of further orientation information. Reference 1, *Battery Design For Aerospace Power Supplies* is a good general article describing manufacturing methods, design, and problem areas. NASA SP-172 *Batteries for Space Power Systems* is also recommended reading.

Three basic elements must be considered in a long-life assurance study of Ni-Cd batteries, viz: the battery itself (design and quality), its usage (charge/discharge cycles), and the environment to which it is subjected (primarily temperature and charge control). Each of these elements is examined in the following text. In addition, the battery analysis is usually divided into four categories to facilitate discussion. These categories are plates, separators, case and electrolyte.

B. GUIDELINES FOR LONG-LIFE ASSURANCE

A shallow depth of discharge, limited overcharge and recharge rates, and low temperature are necessary for a long life battery. The average service life of a Ni-Cd battery can approach five years and/or at least 7000 charge/discharge cycles. Twenty thousand cycles can probably be obtained. These estimates are for

depth of discharge cycles not exceeding 25% of rated capacity and a battery temperature not exceeding 21.1°C. Ni-Cd batteries can be stored for over five years if stored discharged, shorted and about 0°C. Separators are the primary life-limiting elements of the battery.

1. Design Guidelines

1) Design excess capacity into the battery to reduce the percent depth of discharge and compensate for capacity decrease with usage. The penalty is cost and watt-hours/pound;

2) The negative to positive plate area should be at least 1.5:1 so that the negative plate area can absorb the oxygen generated during recharging, preventing battery overpressure;

3) Use non-woven polyproplene separators since they degrade slower than nylon at higher temperatures. The non-woven configuration wets more readily;

4) Hermetically seal the battery to avoid degradation of other spacecraft parts by the electrolyte;

5) Either plate the terminal seal braze with nickel or consider using a nickel-titanium braze material to reduce the probability of electrolyte attacking materials containing copper;

6) Employ a pressure relief valve (200 psi or less setting) for batteries used in manned missions to prevent crew injury in case of battery overpressure. Provide backup monitoring of the battery to terminate/reduce the charge should the primary system fail. Replace battery if pressure relieved because chemical balance is upset and cell case probably is distorted;

7) Use 304 or 304L stainless steel for case and cover material. These materials have proven satisfactory;

8) Use ceramic to metal terminal seals that are more KOH resistant than glass.

2. Process Control Guidelines

1) Employ clean areas during processing and manufacturing to reduce the amount of harmful contaminants. Also, use clean lint-free cotton gloves when handling components. Store components in clean plastic bags when not being processed;

2) Employ clean processes, remove the carbonates and keep the nitrates content down to prevent gas pockets that pop off active material, (See text for recommended limits);

3) Flush plates after KOH is used in the process to form active hydroxides to remove carbonates;

4) Flush and brush plates prior to installation to remove contaminants;

5) Coin plates flat. Flex and clean plates prior to assembly. Have resident inspector examine plates for conformity just prior to cell assembly. These actions will reduce the probability of short by either projection of jagged wire filament through the separator or loose particles of plate material or sometimes tab failures;

6) Weigh each plate to be certain weights are within ±3½% of mean. Also, perform actual capacitance measurements to check plate matching. Mismatched cells can prevent full battery charge;

7) Control the brazing temperature-time relationship to prevent excess dwell during brazing operations that can cause active material penetration of ceramic seals;

8) Avoid rapid cooling after brazing to prevent cracked ceramics and brazing voids.

9) Purge cells of air prior to injecting electrolyte to prevent KOH reacting with CO_2 to form carbonates;

10) Place plates under serialized control and provide traceability for separators and electrolyte material to improve the quality of individual cells which has varied more than desired;

11) Require process and test controls for each active element--plates, separators and electrolyte to reduce end product variability.

3. <u>Test Guidelines</u>

1) Helium leak check the assembled cells. Option-chemical leak check with phenolphthalein;

2) Subject battery during acceptance test to a minimum of three charge/discharge cycles, high impedance short test, and leakage tests. These tests should provide assurance that the basic operating characteristics and construction are satisfactory;

3) X-ray along three axes to find gross battery defects;

4) Conduct a minimum of 30 charge/discharge cycles on assembled cells to eliminate infant mortality and to confirm the matching of individual cells. Resident inspection should observe and confirm these tests.

4. <u>Application Guidelines</u>

1) Maintain battery within a -20°C to +22°C temperature range to retard separator deterioration;

2) Keep depth of discharge below 25% of rated capacity to assure a longer life;

3) Limit recharge and overcharge as denoted below to assure longer battery life.

   a) The recharging rates should be limited to the range of C/2 to C/10. (C = rated capacity in ampere-hours)

   b) The overcharge should be limited to:

   105% C @ 0°C
   115% C @ 25°C
   125% C @ 40°C

4) Plan to replace batteries operating under favorable usage and environments every five years (if feasible) - their maximum estimated life.

5) Store Ni-Cd batteries discharged, shorted and about 0°C to obtain a storage life of about five years.

6) Monitor individual cell voltages for indications of cell deterioration and potential replacement requirements.

7) Erase most memory by discharging battery, short for 16 hours, and then recharge if application permits and need arises. Repeated shallow depths of discharge can prevent future fuller depths of discharge ("memory").

C. LIFE LIMITING PROBLEMS AND SOLUTIONS

The three main areas of consideration are the battery itself, its usage, and the environment to which it is subjected. The usage and environmental effects are examined following the analysis and discussion of cell failure modes. Battery life is particularly affected by the depth of discharge and temperature. Recharging must be carefully controlled to assure a full charge (if required) and avoid overpressure.

1. Failure Mechanism Analysis (FMA)

Failure of a battery can mean several things. It can be a physical failure (i.e., seal leakage, tab breakage) or an electrical failure (capacity or voltage anomalies). Any factor in the battery that causes it to cease functioning as is required by the system would be described as a life-limiting factor. Table 2 presents a summary of the failure mechanism analysis. It delineates the failure modes of the sundry battery elements, their effects upon the battery, the probable failure mechanisms and recommendations on how to eliminate/minimize the failure modes. The following paragraphs discuss and elaborate upon the failure modes and mechanisms. They also recommend means to delete/alleviate the failure mechanisms to enhance the probability of obtaining a reliable long-life battery. For convenience, the discussion is divided into plates, separators, case and electrolyte.

a. Plates - The active materials are cadmium hydroxide for the negative electrode and nickel hydroxide for the positive electrode. The active material is impregnated on plaques that have been coined flat and smooth. Most plaques are composed of either nickel screen or perforated nickel sheets upon which carbonyl nickel powder have been sintered. General Electric cell plates are perforated steel sheets which are dipped in a slurry. The loss of active material will decrease the capacity of the battery.

The loss of the active material may be either due to actual physical loss of material or permanent passivation where negative plate material becomes chemically inert. The migration of cadmium is not normally a problem except for long duration missions when the separators degrade. One obvious solution is to build excess capacity (larger plate areas) into the plates so that the required capacity is still available at the end of the mission; cost and weight constraints may prevent this solution.

Table 2 Failure Mechanism Analysis - Nickel Cadmium Batteries

Part and Function	Failure Mode	Effect on Battery Output	Rel. Rank	Failure Mechanisms	How to Eliminate/Minimize Failure Mode
A. Plates (Contain charge)	Loss of active material	Lessens capacity available	2	1. Permanent passivation 2. Shedding 3. Redistribution or migration of Cd	1. Operate within 0 to 22°C range. 2. Use proper plate geometry for greater heat dissipation. 3. Don't overcharge excessively. 4. Employ clean processes, remove nitrates and keep carbonate content down to prevent gas pockets from forming underneath that pops off material. 5. Provide excess of cadmium oxide. 6. Start with battery with excess capacity, penalties permitting.
	Short	Lower capacity Lowers voltage High temperatures		1. Plate tabs broken, burned or shortened against case or other plate 2. Plate buckling 3. Projections of jagged wire filaments penetrates separators. 4. Loose particles of plate material or metallic particales introduced during processing. 5 Mechanical environments.	1. Don't weld tabs on - make part of substrate. Use wider tabs. Option: coin plates to receive welded tab. 2. Coin plates including all four edges, smooth. 3. X-ray for misalignment determination. 4. Employ clean processes and materials. Flex and brush off plates just prior to assembly.
	Plate mis-matches	Capacity decreased		1. Active material applied uneven or wt. out of tolerance.	1. Require wt. of plates to be within ±3½ of that required.
	Memory	Capacity available limited.		1. Temporary passivation. 2. Depressed operating voltage	1. Completely discharge, short, and recharge to wipe out most of memory.
	Contaminates	Lower voltage & current		Carbonate contaminates in plates.	1. Brush and flush plates prior to sealing cells.
B. Separators (separate, insulate, absorb, and conducts)	Low resistance	Capacity decrease	1	Separator deterioration including dissolved, burned, pinpoint penetration, and impregnated with negative plate material.	1. Limit operating temp. range of battery to 0 to 22°C; 0°C preferred. 2. Use alkali resistant material such as polyproplene or nylon. 3. Strict material and process controls. 4. Perform insulation resistance tests on material.
	Contaminates	Lower voltage & current		Material deteriorates, carbonates formed.	1. Use polyproplene for long-life applications. 2. Low battery temps (0°C) retards deterioration.

Table 2 (concl)

Part and Function	Failure Mode	Effect on Battery Output	Rel. Rank	Failure Mechanisms	How to Eliminate/Minimize Failure Mode
	Poor KOH absorption and distribution	Higher temperatures. Lower capacity over charging. High voltage on charge and low voltage on discharge	1.	Improper material and weave configuration.	1. Don't use woven nylon. 2. No non-woven configurations except for nylon material. (Polyproplene more difficult to wet than nylon.)
C. Case (Contain and support)	Leak/ burst	Lower capacity, eventually becoming an open circuit.	3	1. Oxygen overpressure due to overcharging. 2. Seal or weld leakage or failure. 3. KOH-case material not compatible. 4. Under designed structure	1. Employ high pressure relief valve/ burst disc for manned mission. 2. Limit overcharge, especially above 80% full charge (third electrode, coolometer, voltage limit, thermistor, stabistor or 2-step regulator). 3. Proper ratio of negative to positive plate capacity. 4. Proper quantity of electrolyte-just enough to wet plates and separator. 5. Leak test assembled cell. 6. Proper process control. Weld per MIL-W-8611A. Passivate per MIL-F-14072, finish 300. 7. Use 304L, cond. A per QQS - 766 or equiv. 8. Ceramic-to-metal seal preferred. Suggest stress relieving design such as a "floating" seal. Consider redundant sealing surfaces.
	Post to cell cover short	Loss of capacity, heating		1. Ceramic failure 2. Electro-metallic bridging across ceramic	1. Minimize quantity of braze used with attention given to its elimination on interior side.
D. Electrolyte	Freeze	No output.	5	1. Low temperatures.	1. Keep storage temp. above - 48°C.
	Contaminate			2. Carbonate & nitrate contaminates	2. Limit carbonate and nitrate concentrations to 0.01 gm/liter and 1 mg/liter or less respectively. 3. Don't expose to air as KOH has infinity for CO_2.
E. Internal Electrical connections (Conduct current)	Open	Partial or complete loss of capacity, voltage.	4	1. Mechanical breakage of cell terminals, plate lugs or welded joints.	1. Strict QC. 2. Avoid overly severe dynamic stresses during usage.

The material may be popped-off by gas pockets formed underneath due to carbonate and nitrate contaminates. Elevated temperatures and thermal cycling during charging and discharging can also cause loss of material because of thermal expansion and contraction of this very porous material. The formation of gas pockets can be emended if the manufacturer maintains clean processes, removes the carbonate and keeps the nitrate content down. Hence, the procurement specification should require a clean work area and require written material procurement and process controls that will limit/reduce the amount of nitrate, carbonate and other contaminates. The consensus of opinion among users is that the nitrate and carbonate content of electrodes should not exceed, respectively, 330 micrograms and 10 milligrams per gram of the active material, sinter and substrate. The adverse effects (shedding) of thermal cycling and elevated temperatures can be alleviated primarily by maintaining the battery within a narrow temperature range; the temperature should be kept within 0° to 22°C - a range that enhances battery life as discussed later in the hardware life section. Proper plate geometry and battery case design can aid in heat dissipation and minimal temperature gradients across the battery.

Internal battery shorts are usually caused by two mechanisms: Projections of jagged wire filaments thru the separators and/or loose particles of plate material or metallic contaminats. A third possibility is either positive plates or tabs touching the negative case or positive and negative plates or tabs touching depending on configuration. The plates are normally coined to prevent protrusion of wire filaments. However, according to Mr. Mains of NAD-Crane, Indiana, not all manufacturers coin plates. In addition some manufacturers produce and coin only one size of plate for economy reasons; hence, when a smaller plate is required, the larger standard and coined plate is cut, leaving two rough edges. Procurement specifications should require all plates to be coined and all edges to be smooth. Coining is the compacting of the edges of the plaques under high pressure. An optional approach is to cover the edges with an long-life epoxy coating. However, epoxy coating has been known to shed and also to have affected electrical performance.

Loose plate material can be removed by carefully flexing and brushing-off plates just prior to assembly. Potential shorts due to misalignment or poor workmanship can sometimes be detected by radiography. Although radiography of the assembled battery normally shows only gross defects, it should be required. Loose plate material is very difficult to detect since only about 5% of the battery volume is outside the projected area of the plates.

Copper contaminates can react with KOH and burn holes in the separator material, producing a telltale blue spot and a subsequent short. The copper is usually from spot welding operations; vigorous control of spot welding can overcome this problem.

Tabs should be as wide as configuration and weight constraints permit. Wide tabs provide additional strength and current and thermal conductivity. The tab should be an integral part of the plate to avoid welding it to the substrate. The plates should be coined to receive the tab if a welded configuration is necessary; also if the tab weld is not sufficiently cleaned, heating problems can occur.

If the plates are not closely matched in capacitance, one cell of a battery will become fully charged before the others. The end result is that the cells in a battery are at various stages of charge and the battery does not obtain full capacity. According to Mr. Leuthard of Martin Marietta Aerospace, mismatch is magnified with time, i.e., battery capacity decreases with time. Since capacity is a function of the mass of active material, controlling plate weight will provide plate matching within required limits. The consensus of those interviewed indicated that plate weights should be controlled within $\pm 3\frac{1}{2}\%$ of the mean. Procurement specifications should require this weight matching. In addition, actual capacitance measurements should be taken during cycling testing (to be discussed later) to check cell matching.

A Ni-Cd battery can develop a "memory" if it is subjected to repeated shallow depths of discharge. The battery cannot provide deep discharges because of temporary passivation. The "memory" can be mostly erased by discharging the battery, shorting for say 16 hours, and then recharging.

According to Mr. Ford of NASA/Goddard, carbonate contaminates effect the rather delicate chemical balance in a cell, poisoning the negative electrode. Nitrates are also undesirable. These contaminates will eventually effect cell voltage, especially at lower temperatures. The plates should be flushed and brushed prior to installation to remove contaminates.

b. Separators - In addition to providing mechanical separation of the positive and negative plates, separators must be able to absorb and retain sufficient electrolyte to provide a conducting path between the plates. They require a high electrical resistance and a resistance to strong alkalis in a strong oxidizing atmosphere. In addition, they require a gas permeability sufficient to allow the oxygen produced at the positive during overcharge to reach the negative.

Separators are the primary life-limiting elements of the battery. The separator material deteriorates with time and temperature until they no longer either adequately separate the plates mechanically or provide sufficient electrical resistance. Undesirable carbonates are sometimes byproducts of their deterioration. Nylon has been satisfactorily employed for years as the separator material; however, polyproplene is more oxidation and temperature resistance than nylon. Polyproplene is difficult to adequately wet and should be used only in a non-woven configuration. It was the consensus of the literature and those interviewed that polyproplene should be used for long-life and/or higher temperature applications. The life of either nylon or polyproplene is extended if the temperature is kept below 22°C, preferably around 0°C.

c. Case - Leak or rupture are the cover and case failure modes. Leakage usually occurs in only four places: weld of cell cover to case, negative tab welds to case, and at the insulating seal used to separate the positive terminal from the case and the pinch-off tube. The case and cover is normally made of 304L stainless steel, condition A per MIL-W-8411 with all welded areas passivated per MIL-F-14072, finish E300. This case material has proven adequate. The case is sometimes 304 because of the cost and short supply of 304L.

Ceramic seals (alumina of 99.4% purity) are almost universally used today. They can withstand thermal shock better than the glass seals that were once prominent. The KOH attacks glass. The ceramic-to-metal seals were originally developed by electron tube manufacturers. Figures 3 and 4 show two improved ceramic-to-metal terminal seals. Note that single failure of the braze to the cover will allow leakage.

The braze area should be as large as possible to increase the sealing surface area. Stress relief members are employed to decrease the adverse effects due to thermal expansion or plate movement during charging/discharging and dynamics. Weld and seals are usually helium leak checked after assembly; this should be part of the procurement specification.

The KOH electrolyte can attack the silver-copper-platinum braze material that has been predominately used. A nickel platting has been employed over the braze to protect it from the KOH. About

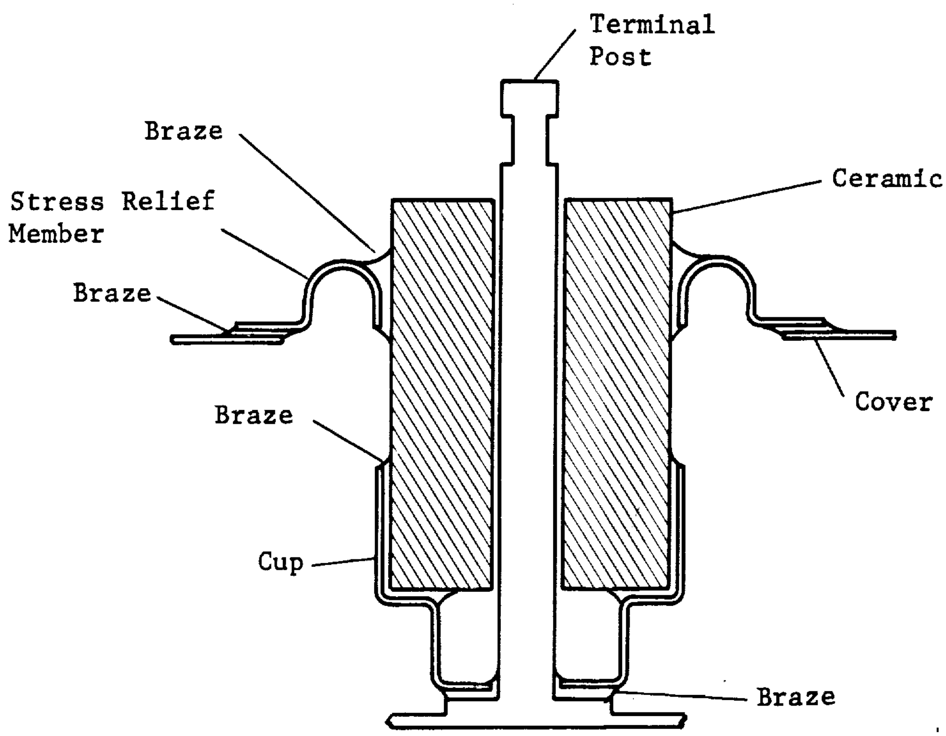

Figure 3 Ceramic-to-Metal Terminal Seal

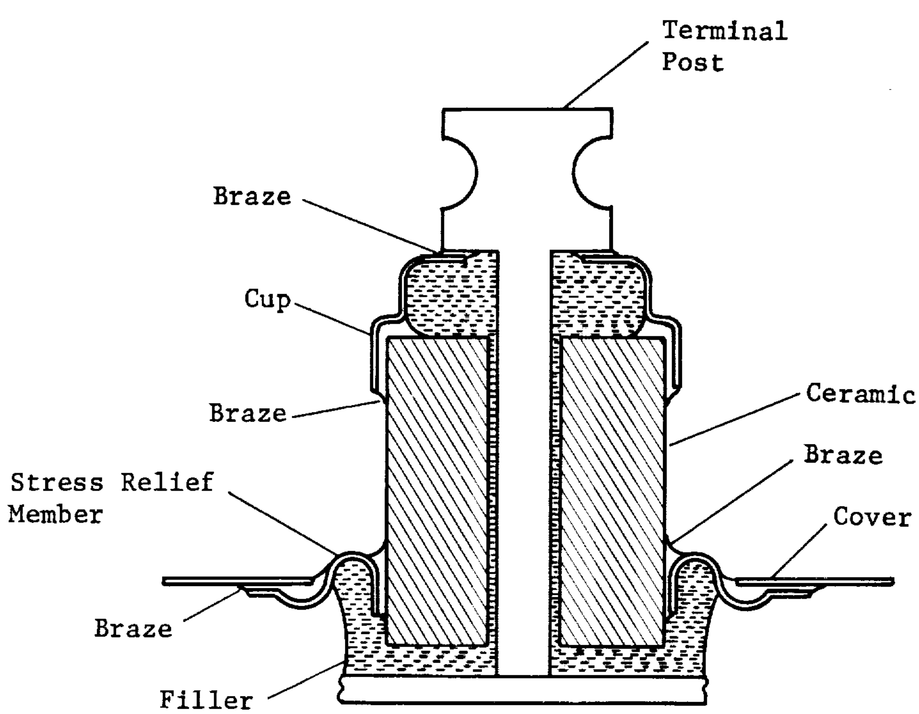

Figure 4 Ceramic-to-Metal Terminal Seal

four years ago, General Electric (GE) developed a nickel-titanium braze material that is not attacked by the electrolyte. The GE seal design also incorporates long leak paths. In four years of testing, two at GE and two at NAD Crane, no failures have appeared. The GE seal should be given consideration when selecting a seal design.

High pressures can be developed during recharge due to oxygen generation. Gross overcharging can also release hydrogen, a dangerous situation should a short occur. Ni-Cd batteries have on isolated occasions exploded. The oxygen build-up begins slightly below 80% of full charge; hence, means to reduce the charge rate above 80% should be employed. The application guidelines subsection discusses charging methodology in some detail. For manned missions, pressure relief valves on sealed Ni-Cd batteries are mandatory. The pressure setting should be well above the "normal" overpressure of about 50 psia to prevent leakage. A relief pressure at the lesser of the proof pressure or about 200 psi is suggested; most batteries can withstand this pressure with only bowing (not rupture). Should a valve relieve, KOH and air could form carbonates on the valve, perhaps failing the valve open or closed. Should a cell relieve, it should be replaced since the chemical balance is upset. Perhaps a burst disk could be employed in lieu of a relief valve. The open burst disc would give a positive indication if a large overpressure had occurred and the causes investigated. Oxygen overpressure in a Ni-Cd battery will decrease to normal operating pressures after 10-20 hours of nonoperation if the battery is functioning properly (oxygen absorbed); this fact can be used as a check on battery condition. Hydrogen overpressure will also disappear, but it takes about six months.

d. Electrolyte - The electrolyte is normally a 34% potassium hydroxide (KOH) solution. KOH has an affinity for the carbon dioxide in the atmosphere, forming carbonates. This is why the plates must be flushed after KOH is used in the process to form active hydroxides. Batteries should be purged of air prior to injecting the electrolyte thru fill tubes; a purge gas mixture of 95% oxygen and 5% helium at 5 psig is suggested. The presence of helium allows for subsequent helium leak checks. Each fill tube should be pinched closed and welded shortly after filling. Contamination of KOH should be prevented by utilizing burets while filling to minimize exposure to atmospheric conditions.

e. Failure Detection Methods - Most internal shorts can be determined by a stand test, a capacity test, and a vibration test. The stand test consists of completely discharging the battery, allowing it to stand about 16 hours at a controlled temperature, then measuring the cell voltage. The cell voltage at the end of the stand period can indicate the existance of an internal short. The standard electrical capacity test, which determines the number of ampere-hours available from a cell under fixed current and temperature conditions, can also be used as an indicator of internal shorts or high internal resistance. The vibration test consists of vibrating cells before and after battery assembly. Many irregularities are indicated from monitoring the voltage via an oscilloscope before, during, and after vibration.

Radiography of the assembled cell can sometimes detect loose material and gross workmanship defects that could cause shorts. It is suggested that neutron radiography be investigated as an inspection tool. In many applications, neutron radiography is replacing the conventional X-ray because much more detail is available in neutron radiography photographs.

Seal integrity can be checked by: (1) performing a helium leak test prior to cell assembly at the proof pressure is suggested since it also acts as a seal proof pressure test, and (2) subject cells to a chemical-indicator leak test after assembly and again after all testing.

2. Design

Much of the design criteria has been either stated or implied in the FMA discussion. The following selection criteria concerns the battery only. The hardware life and applications subsection discussion covers the effects of usage and environment upon battery life and design. Environments and usage determine battery life about as much as the design of the battery itself.

a. Selection Criteria - Table 3 summarizes the more pertinent design factors to be considered in selecting Ni-Cd batteries for long life missions. Much background information on battery lives, failures, design, test, manufacturing and actual or potential solutions is contained in References 2-7.

The NASA Battery Test Facility, NAD Crane, has conducted cycle life programs since 1963. Over 1500 Ni-Cd batteries have been tested and life parameters analyzed. The NASA battery workshops transcrips provide technical discussions of Ni-Cd batteries. A

Table 3 Design Factors for Long-Life Assurance of Nickel Cadmium Batteries

Design Factors	Remarks
1. Capacity Decrease with Usage	Start with greater capacity than initially required so that required capacity will be available at end of service life. A weight-resources tradeoff is required to determine whether replacement or greater initial capacity is optimum. This trade-off can be performed only for each specific program.
2. Oxygen Overpressure incurred during overcharging	The oxygen is normally absorbed by the negative plates; therefore the negative plate area should be large enough to absorb the oxygen during normal over-charging. A minimum negative to positive plate ratio of about 1.5:1 is suggested.
3. Separator deterioration	The concensus of opinion of those surveyed was that polyproplene separation should be specified for long duration usage. A non-woven configuration has better wetting characteristics and should be specified. Keep the battery temperature below 20°C to retard degradation; battery cooling may be necessary.
4. Hermatically Seal	It is necessary to hermetically seal spacecraft cells to avoid degradation of other parts of the spacecraft due to chemical action of gases from the battery. The quality and stability of the ceramic-to-metal seals are quite good and will handle normal pressures.
5. Large Temperature Gradiants	About 0°C is the optimum temperature for long-life. To avoid hot spots or large temperature gradients, the battery geometry must be correct to maximize heat transfer characteristics. Specify a maximum temperature gradient to be ±5°C cell to cell.
6. Electrolyte attack of terminal seal	KOH attacks brazing materials containing copper. Either plate the braze with nickel or consider using a nickel-titanium braze material. Leak paths should be as long as practical.

soon to be released NASA Goodard specification for Aerospace Nickel-Cadmium storage Cells by Baer and Ford (S-716-P-28) will establish requirements for hermetically sealed Ni-Cd secondary cells with dual ceramic seals. A survey was taken of manufactures and users by Martin Marietta Aerospace to determine life limiting factors and solutions.

b. Survey Results - Two surveys were conducted. One survey involved general questions concerning Ni-Cd batteries; the consensus of the answers is presented in Table 4. The second survey determined why specific batteries obtained long lives and high reliabilities. The results of this survey are summarized in Table 5. Response from both surveys are incorporated in the discussions throughout this chapter.

c. Alternate Approaches - Two general alternatives to the Ni-Cd battery are the Ag-Zn and Ag-Cd batteries. Trade-off studies are necessary to optimize for specific missions. While the Ni-Cd battery is the best of the three from a general and long-life standpoint, the other two, with their superior watt-hour per pound, may be the choice in specific applications. The silver zinc battery may be the logical choice for a shuttle type mission because:

1) Weight of an Ag-Zn battery is about 1/5 that of a Ni-Cd battery for the same capacity;

2) The average price of an Ag-Zn cell is about $210.00 (165-amp-hour), while that of a Ni-Cd cell is about $270.00 (300-amp-hour);

3) The Ag-Zn battery is more vibration resistant than the Ni-Cd battery. The plates can be restrained at bottom and top corners.

It is recommended that the present and projected state-of-the-art for Ag-Zn and Ag-Cd be investigated so that an accurate trade-off can be performed with Ni-Cd batteries for specific applications such as shuttle. For example, a specific battery selection study by Martin Marietta Aerospace indicated that the Ag-Zn battery is a better choice than the Ni-Cd battery for space missions to 500 days on a watt-hour per pound basis.

Table 4 Results of General Manufacturing/Agency Survey - Nickel Cadmium Batteries

Questions	Consensus Answer
1. What is the maximum expected life using low (to 30%) depths of discharge @ 70°F. Cycles? Calendar life? Storage?	Five year life and over 7,000 cycles presently possible. Ten year life may be possible under favorable discharge, temp., and charging conditions. Can store for long periods if cool and shorted out.
2. What is the most probable life limiting failure mode? Cause? Remedial action?	Plate-to-plate short caused by separator deterioration. Use polyproplene separators.
3. What means do you recommend for charge control? Third electrode, other?	Use time-charge relationship with voltage temperature compensation for higher charge rates.
4. What are the adverse effects of carbonate contaminate? Beneficial effects? How can one control/reduce carbonate content.	Carbonates depress voltage levels at low temps. Effects of carbonates not fully known or confirmed. Control carbonate content in plates; wash plates, don't let KOH see atmosphere. Avoid nylon separators for long life applications.
5. What separator material and design criteria is applicable to long life?	Use polyproplene for long life applications. Use non-woven configurations. Keep battery temp low (20°C) to retard degradation.
6. What seal and case material and design criteria do you recommend as applicable to long life?	Ceramic to metal seal preferred. 304 L cover and 304 case material is satisfactory.
7. What plate material, design and controls are recommended for long life assurance?	a. Tabs should be part of substrate; avoid welding on. Tabs should be wide for strength and current carrying capability. Full width tabs desirable, wt. limitations permitting. b. Plates should be matched in weight to within 3½% of mean. Check for capacitance match during cycling tests. c. Grid should be coined to decrease probability of separator penetration. All edges must be smooth.
8. What process controls related to long life are required?	Must control *each* element that goes into a cell. Cleanliness must be controlled.

Table 4 (concl)

9. What inpections and tests do you recommend? Any accelerated testing used?	a. X-Ray assembled battery. It will show gross defects. b. Perform sufficient cycling tests to "burn-in", eliminate infant mortality, and confirm matching and capacity of individual cells.
10. Are there any safety hazards involved with subject batteries.	Battery can burst from overpressure due to rapid overcharging, but probability of occurrence is very low. Can get hydrogen generation if *very* gross overcharging occurs. Employ a backup charge control to prevent gross overcharging. Employ a pressure relief valve for manned missions. Set relief pressure high ($\sim$ 200 psi) to prevent venting unless battery is grossly overpressure. Avoid overdischarging or driving the cells negative to prevent hydrogen generation.

Table 5 Specific Component Surveys

Part Description and Using Program	Users Opinion of Why Part Was Successful	Manufacturer's Opinion of Why Part Was Successful	Recommended Guidelines, Improvements, Procurement Spec Requirements, Etc.	Relative Impact on Yield & Cost of Incorporating Recommendations
OAO-1 Orbiting Astronomical Observatory 20 Amp-Hour Sealed Nickel Cadmium Battery	<u>Opinion</u> Unsuccessful <u>Criteria</u> 3 days operation with design life of up to one year. <u>Reasons</u> 1. Deficient charge control concept and excessive charge voltage limit. 2. Lack of adequate heat rejection capability.	<u>Opinion</u> Unsuccessful <u>Reasons</u> 1. Defect in charge control hardware not a battery design problem.	<u>Recommendations</u> <u>Guidelines</u> Battery Cell vendors should not be responsible for manufacture of the battery due to difficulty in understanding the total Battery/system requirement. <u>Design Improvements</u> 1. Charge control revised to 3 battery parallel operation with constant voltage, multiple level method. 2. Heat rejection capability doubled using increased heat sink area and louvers. <u>Specifications</u> Change from performance oriented cell spec. to one that also covers cell component in-process requirements.	<u>Yield</u> Improvements resulted in successful operation of OAO-2 <u>Criteria</u> Over 3½ years full time operations. <u>Cost</u> Large
OAO-2 Orbiting Astronomical Observatory 20 Amp-Hour Sealed Nickel-Cadmium Battery	<u>Opinion</u> Successful <u>Criteria</u> Currently operating full time with over 3½ years mission time accumulated. Design life of up to one year. <u>Reasons</u> 1. Improved charge control system. 2. Improved battery thermal control. 3. Improved battery cell specification; 1st usage of hi-rel specification.	<u>Opinion</u> Successful	<u>Recommendations</u> <u>Guidelines</u> 1. Reduce Battery preflight operation to minimum. 2. Space Craft thermal design to limit battery operational temperature to range of 0°C to 20°C. 3. Optimize change control for application. <u>Design Improvements</u> 1. Improve separation material in cell. 2. Improve plate material uniformity and Mfg. <u>Specifications</u> 1. Further refinement of the "High Rel" type cell spec.	<u>Yield</u>: 5 year life goal <u>Cost</u>: small small small moderate moderate moderate

Table 5 (cont)

Part Description and Using Program	User's Opinion of Why Part Was Successful	Manufacturer's Opinion of Why Part Was Successful	Recommended Guidelines Improvements, Procurement Spec Requirements, Etc.	Relative Impact on Yield & Cost of Incorporating Recommendations
OGO Orbiting Geophysical Observatory 12 Amp-Hour Sealed Nickel-Cadmium Battery	<u>Opinion</u> Successful <u>Criteria</u> Design life of one year. Demonstrated mission life in excess of 2 years. <u>Reasons</u> 1. Implementation of improved charge control 2. Use of stringent cell screening tests with selection and matching into batteries.	<u>Opinion</u> Successful <u>Reasons</u> 1. Same as user. 2. Better design of battery base plate to allow increased heat rejection.	<u>Recommendations</u> Same as OAO-2	Same as OAO-2
Intelsat IV 15 Amp-Hour Sealed Nickel-Cadmium Battery.	<u>Opinion</u> Successful <u>Criteria</u> Four vehicles in orbited operation for as long as 1½ years. The design life is 7 years. <u>Reasons</u> 1. Maximum depth-of-discharge limited to 50% 2. Low battery operational temperature (65°F-75°F) maintained. 3. Battery charge controlled by ground command and rate limited to C/15 rate. Capacity restored based on calculated capacity output from previous discharge.	<u>Opinion</u> Successful <u>Reasons</u> 1. Rigorous control of cell manufacturing processes. 2. Comprehensive cell and battery screening tests including Space Craft Flight Acceptance test.	<u>Recommendations</u> <u>Guidelines</u> 1. Use low battery operational temperature. 2. Minimize battery over charge. <u>Design Improvements</u> 1. Incorporate further improvements in cell terminal seals. 2. Investigate further use of new plate separation materials such as types of polypropylene. <u>Specifications</u> 1. Further refined comprehensive cell specifications.	<u>Yield</u>: 10 year life goal <u>Cost</u>: Moderate

Table 5 (concl)

Part Description and Using Program	Users Opinion of Why Part Was Successful	Manufacturer's Opinion of Why Part Was Successful	Recommended Guidelines Improvements, Procurement Spec Requirements, Etc.	Relative Impact on Yield & Cost of Incorporating Recommendations	
				Yields	Cost
Nimbus Vehicles I thru IV 4 Amp-Hour Sealed Nickel-Cadmium Battery.	Opinion Successful Criteria Four vehicles in orbital operation for as long as 3 years. The design life was six months minimum. Reasons 1. Low battery operational temperature ($25°C \pm 10°C$) maintained. 2. Use of stringent cell screening tests.	Opinion Successful Reasons 1. Space Craft power management and charge control limits battery overcharge and results in maintaining energy balance. An auxillary load is used to control power available for battery charge/overcharge 2. Voltage limit charge control with over temperature cut-off employed.	Guidelines 1. Conduct Space Craft integrated tests in space simulated environment to assure system/battery performance. 2. Where possible design thermal control for battery operation at 10°C to 20°C. 3. Incorporate automatic battery over-charge control using signal electrode battery cells. Design Improvements 1. Employ rectangular cells instead of the cylindrical type cells used on Nimbus. 2. Use dual insulated terminal feed through in cell design. Specifications 1. Use of battery handling document that maintains stringent control of battery temperature and usage prior to launch critical to long mission life.	Excess of 3 years	Moderate Small

d. Hardware Life and Application Guidelines - Besides the battery design itself, the battery usage and the environment to which it is subjected have a direct bearing upon battery life. To estimate the life of a battery, it is necessary to specify the maximum depth of discharge and the maximum battery temperature.

A shallow depth of discharge is vital for battery long life. A general concensus of manufacturers recommends under 25% depth of discharge. Figure 5 illustrates how the depth of discharge affects the cyclic life of a typical Ni-Cd cell. It can be noted that depth of discharges greater than 40% of rated capacity greatly decreases cyclic life. Very shallow depths of discharge (10%) greatly enhance cyclic life. Mr. Vogentzie of General Electric Company pointed out that one Sonotone cylindrical battery has functioned for over nine years in space, but the depth of discharge was only 10%. Use of a battery with "excess" capacity is one way of keeping the depth of discharge lower. A cell has both a characteristic end of discharge voltage and a plateau voltage. A cell should be discharged to their plateau voltage but not beyond for long life. Incidentially, the lower the rate of discharge, the greater the capacity available to an external circuit and the lower the internal temperature.

Temperature is one of the most important life-limiting factors. Many unfavorable effects result from using a battery at other than optimum temperature levels. The separators degrade more rapidly at higher temperatures, especially nylon at above 25°C. The capacity of the battery is substantically decreased at temperatures below -10°C. The concensus of those surveyed believed an operational temperature in the battery of about 0°C to be about optimum - separator and plate degradation is minimal. Also approximately 0°C is the optimum storage and non-operational temperature. The absolute lower temperature limit should be above the freezing temperature where frozen electrolyte could cause separator penetration.

It is apparent that means must be provided to maintain the Ni-Cd battery at about 0°C to ensure a longer service and storage life. NAD, Crane, Indiana, (Ref. 8) arrived at the following conclusions following a cycle life program on 1586 batteries; "Life cycling data shows that nickel-cadmium cells tests at 0°C give longer cycle life, higher end of discharge voltages and less degradation of ampere-hour capacities than cells tested at 25°C or 40°C. Overall performance decreases with increase in the depth of discharge at all test temperatures."

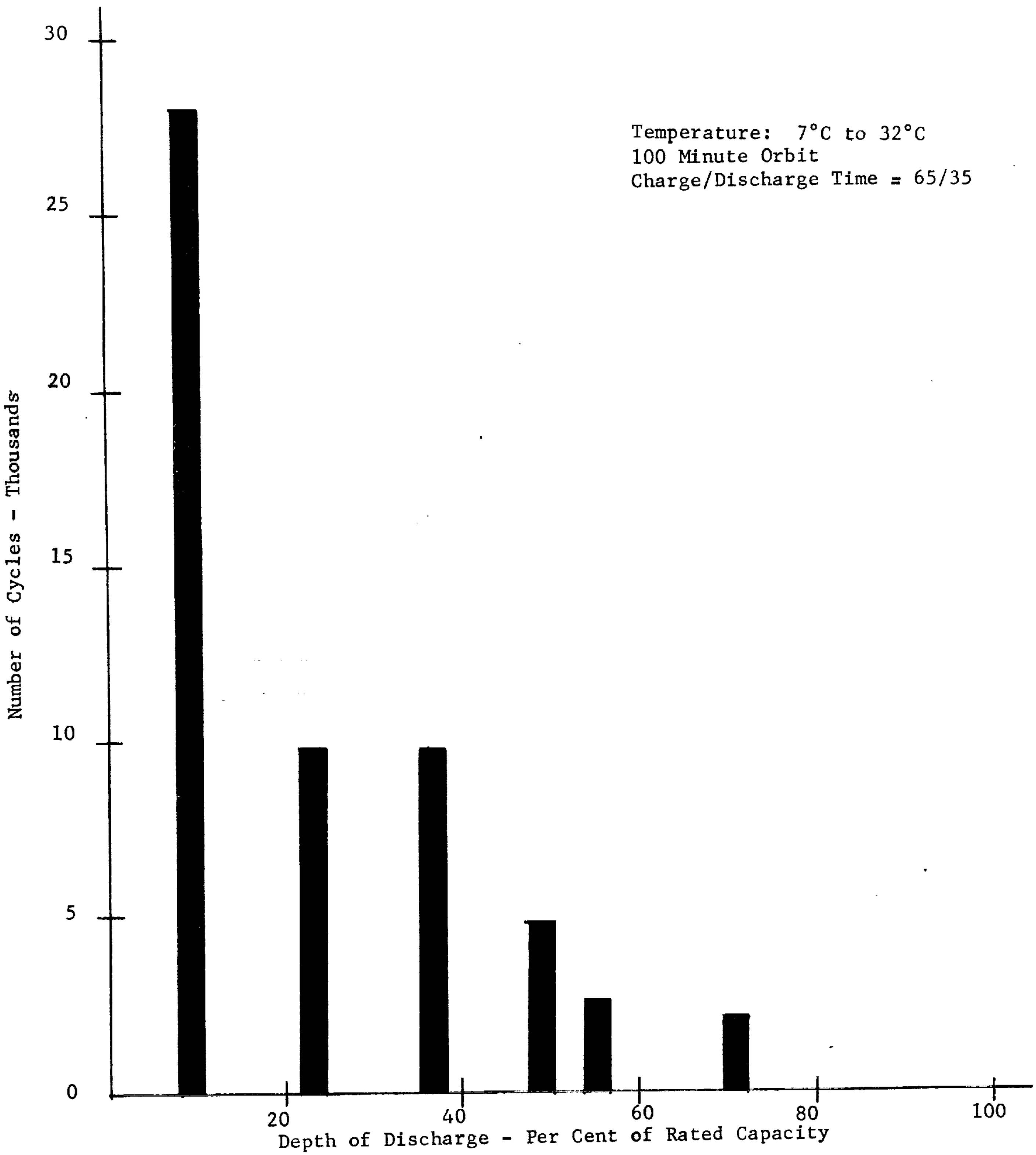

Figure 5 Typical Average Cycle Life of Hermetically Sealed Ni-Cd Cells vs Depth of Discharge

The rate of charge is another parameter to consider for long-life applications. Again quoting reference 8: "Cell cycle life is extended when the amount of recharge is limited to the following amount:

105% C @ 0°C.
115% C @ 20°C
125% C @ 40°C

Operating performance can also be improved by recharging at rates of between C/2 and C/10 with the recharging rates controlled by auxiliary electrodes or cadimum coulometers."

The survey consensus of opinion was that using the present state-of-the-art, a five year service calendar life and over 7000 charge/discharge cycles (with 30% depth of discharge cycles) are possible. These estimated lives assume the temperature does not exceed 22°C. Ni-Cd batteries can be stored for long durations (over 5 years) if discharged, shorted and kept cool (about 0°C). While it is true that a few batteries have functioned longer than five years and 7000 cycles (one 12 amp-hr battery went 40,000 cycles @ 0°C), it is believed that one should anticipate replacement of Ni-Cd batteries for manned missions at about five years. Some of those interviewed believed that a ten year life may be feasible, but demonstrating this long a life (accelerated testing) was not currently practicable. It was suggested that real-time half-life life tests of the best of current batteries commence now to ascertain their lines under optimum conditions.

The probability of a battery failing catastrophicly (suddenly) is remote; they degrade with time and usage. Thus a review of battery instrumentation data can indicate the degradation of a battery and the need for replacement. The voltages in each cell (or a random selection) should be monitored before and after recharging for performance analysis. Mr. Schulman, Gulton Industries, suggested that batteries in storage be recharged and checked every 3-4 months to check for any indication of degradation.

At this point, some discussion on charge control methodology may be in order. The charge control methods used are voltage limit control, auxiliary electrode, coulometer, stabistor, a two step regulator, thermistor controlled voltage limit and the Sherfey upside-down cycling regime. A discussion of charge control methodology is not within the scope of this study; however, the auxiliary electrode and the voltage limit control are briefly discussed for orientation purposes. These two are the most commonly employed methods.

Nickel-cadmium cells have been developed with an auxiliary electrode whose voltage, with respect to the negative terminal, is dependent upon the partial pressure of oxygen in the cell. When a nickel-cadmium cell is being charged, it generates oxygen very slowly until it nears 80 percent of the required recharge; then suddenly, the amount of oxygen generated internally increases rapidly. The increased oxygen pressure causes a fast rise in voltage between the auxiliary electrode and the negative terminal. This increasing voltage is used to signal a control circuit to reduce or terminate the charge current. The charge-current control circuit utilizes the auxiliary electrode voltage on selected cells in the pack to reduce the charging rate after the cells have received the desired amount of recharge. The circuit is designed to monitor the auxiliary electrode voltage of each cell while it is being charged. As the auxiliary electrode voltage of any one cell of the pack approaches a preset value, the circuit begins to reduce the charge current. When the auxiliary electrode voltage of any cell reaches the predetermined voltage (trip voltage), the charge current will be reduced to a preset trickle or to zero. The major disadvantage is that the oxygen pressure is affected by temperature. The output current for a given capacity will vary somewhat with temperature, requiring temperature sensing and a related output control circuits. Also because of tolerances, the voltage outputs may vary from cell-to-cell. Mismatch of cells (capacity) could prevent the battery from receiving a full charge since some cells may reach full charge first, calling for a charge rate decrease.

In the conventional two-step charge control method, cell packs are recharged using a pack voltage limit. All charging is constant current until the voltage limit is reached; at this time the charge current is automatically reduced to protect the cells during overcharge. The charge current is determined by the theoretical percent of recharge needed to be returned to the cells following a known amount of discharge. The capacity returned for a given voltage is temperature dependent. Figure 6 shows a typical cell voltage VS capacity returned. The curves assume a given prior depth of discharge and a specified following charge rate.

It is not possible to say that one charge system is superior. The mission constraints and usage can affect selection. The following conclusions may aid in charging method selection for denoted parameters. The conclusions are from *Parametric Charge*

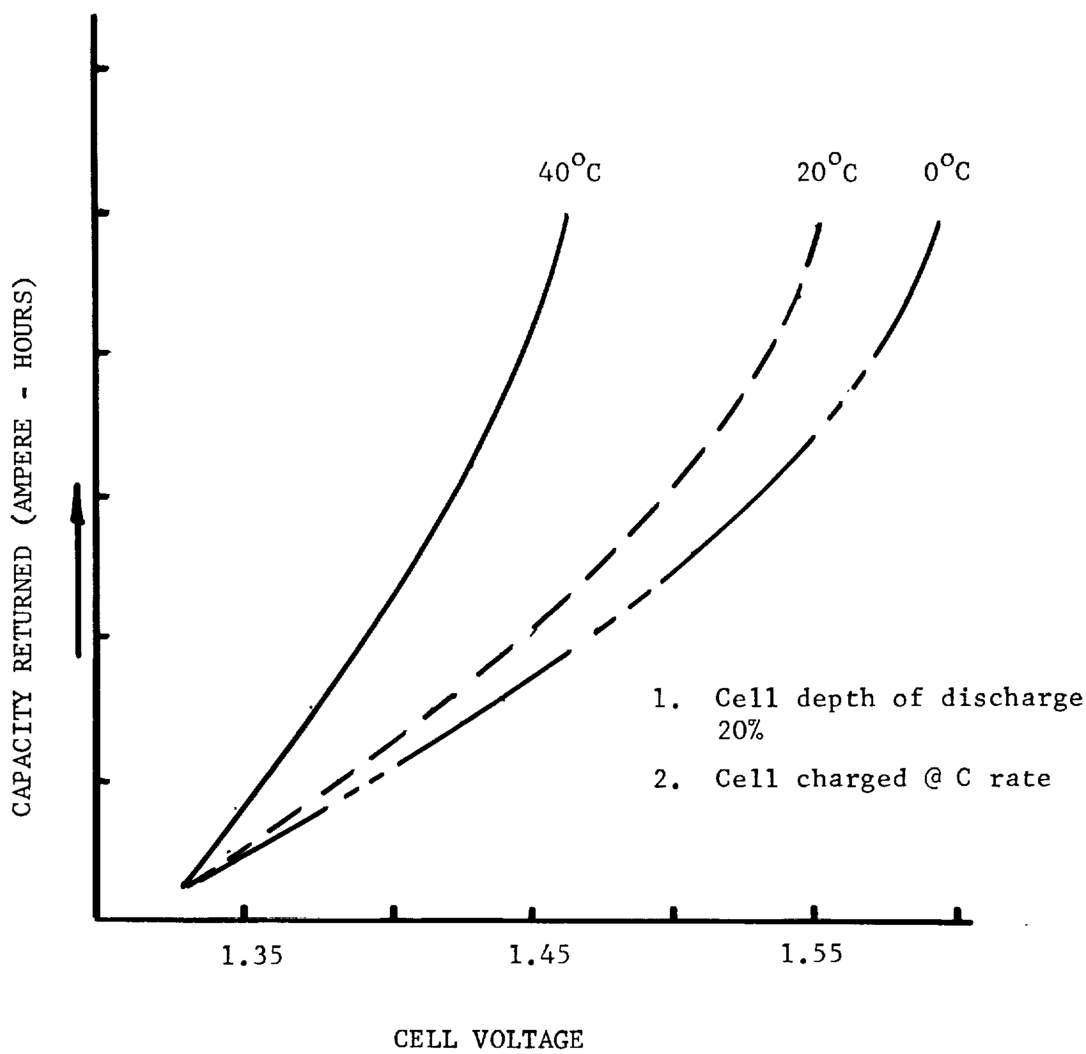

Figure 6 Capacity Returned vs Cell Voltage

Studies for Aerospace Nickel-Cadmium Batteries by Prusse, Shain, Betz and Sylvia of Gulton Industries (Ref. 10). The conclusions are for near earth orbits with 65 minutes of charging and 35 minutes of discharging.

1) Initial charge voltage depends on state-of-charge, and does not depend on temperature;

2) The conventional two-step charge control method is a successful means of recharging cells previously cycled to 20% depth of discharge. It cannot be successfully employed to recharge cells previously cycled to 40% depth of discharge;

3) The auxiliary electrode charge control concept at the C or C/2 charge rate is a successful means of recharging cells previously cycled to 40% depth of discharge. At the C rate, a 60% maximum depth of discharge can be recharged using the auxiliary electrode.

D. TEST METHODOLOGY AND REQUIREMENTS

1. Qualification Tests

Demonstrating a Ni-Cd battery life of long duration (to ten years) is not currently possible. Continuous and more rapid cycling can demonstrate the necessary cyclic life in some instances; however, accelerated testing methodology to demonstrate calendar aging has not been successful. The industry survey supported this conclusion. The most common accelerated testing technique is to subject a test specimen to elevated temperatures for relatively short periods of time and correlating the failures to real time and normal operating temperatures. Rapid cycling and/or deep depth of discharge can also accelerate aging (raising internal temperatures). Unfortunately some of the battery failure modes are temperature dependent. For example, the nylon separators fail rapidly above 50°C, but are stable around 0°C.

The calendar life of a battery can best be estimated by determining the lives of existing batteries with very similar construction and materials. Probably the best source of such data is from NAD Crane who have been performing accelerated battery life tests

since 1963. NAD Crane has been attempting to develop a life prediction model when applied to an accelerated life situation. This effort is described in QE/C 70/787. The mean and variance of the Bayesian posterior distribution was used in the prediction model.

The battery qualification tests simulating anticipated shuttle mission environments will be difficult because of the 100 launch and re-entry environments encountered during the life time of each spacecraft. The cumulative time for each environment can become relatively large. At this time only cursory estimates can be made of some of the shuttle environments. It has been estimated that some components in the most adverse vibration locations could see 30 to 50g rms random for 1 3/4 hours total. If a safety factor of four is selected, a component such as a battery might be subjected to as much as 7 hours of vibration per axes during a qualification test. According to Mr. Mains of NAD Crane, cells have successfully withstood up to 70 g rms vibration for five minutes. Whether a battery could withstand up to 7 hours per axis at these vibration levels is not known. The fatigue characteristics and limitations have not been fully defined. The tabs and plate structure may be prime candidates for early fatigue failure since the plates are not completely anchored solidly inside the battery.

2. Screening and Burn-In Tests

The majority of the aerospace battery users interviewed believed that the quality of the individual cell components varied more than desired. More consistent and longer battery lives would result from more comprehensive and stricter process controls and inspection screening tests. Most of the battery manufacturers also manufacture commercial batteries where competition forces certain economies. Greater controls for aerospace batteries will mean greater costs, but better batteries. Use of the tentative NASA/Goddard battery specification (S-761-P-6) should alleviate this situation to some extent.

Tests should be conducted on the individual active components of a battery on a lot sampling basis to insure constant life characteristics. The "active" components are the plates, separators and electrolyte. The life of a battery is limited by the life of the battery component with the shortest life (separators). However, should the lives of other components decrease below that of the separators, the battery life is reduced. Hence it is vital that tests confirm the quality of components, The test methodology is known and need not be herein described. The procurement specification should specify these tests in such detail that consistent high quality is maintained. Prevention of the failure modes discussed earlier should be considered in the test specifications.

The assembled cell should be subjected to sufficient burn-in to eliminate infant mortality and confirm the matching capability of individual cells. The burn-in is accomplished by a number of charge/discharge cycles at a specified temperature, usually about 25°C and 0°C. The cycle frequency and depth of discharge are varied to check various performance characteristics such as capacity, pressure, cell voltage, recovery, etc. Some of the cycles should be the same as its intended spacecraft usage. All vital parameters should be measured; automatic cycling and recording equipment is normally used. Comparison of individual cell voltages provides a good indication of a cell's performance and matching. A cell with a voltage lower than the others should be rejected. A minimum of 30 charge/discharge cycles is recommended to provide some statistical significance of results. Resident and trained inspectors should observe and confirm these tests.

The following minimum acceptance test is recommended:

1) Three overcharge/discharge cycles to see if required capacitance is obtained. The battery temperature should be at the actual/anticipated mission operating temperature;

2) One short test to check for high impedance shorts. Discharge to 0 volts, short with a 1/2 ohm resistor for 16 hours, and allow to set open circuit. The voltage should return to greater than 1.15 volts;

3) Subject test specimen to vibration (may be combined with the three charge/discharge cycle tests). X-ray test specimen before and after the vibration tests to check configuration, workmanship, and particulate matter (Note: Temperature cycling tests are not possible because of possible high temperature degradation.);

4) Check for case and seal leakage with either hydrogen leak testor or chemical alkali tests (phenolphthalein) under vacuum.

E. PROCESS CONTROL REQUIREMENTS

Until recently many aerospace cells were procured by specifying the output characteristics and the qualification tests. Because of the the greater reliability and longer lives required for space applications, the users are now beginning to look inside the cell and attempt to control the quality of the components.

A small variation in a process can change the life and reliability characteristics. For example, the quantity and distribution of KOH electrolyte is important. Insufficient KOH and some areas can be dry. Too much electrolyte can prevent proper oxygen recombination or the third electrode (if used) doesn't function properly. If the electrolyte is drained from a cell, only a few cc would be obtained, the rest remaining in the separators and electrodes. (Ni-Cd batteries are essentially "dry" or "starved.")

The consensus of those interviewed (Table 4) was that each component that goes into a cell must be controlled to assure a good end product. Strict control of existing processes is the key to uniformity. The vendor should be required to supply a Manufacturing Control Document (MCD) to the customer. The MCD identifies the manufacturing processes, procedures and inspection documents in the production of parts. The users should use discretion when specifying process controls that might require altering a vendor's process. Mr. Voyentzie (General Electric) pointed out the problem of drastically changing process controls that have been working for years. It may take years to confirm the new processes. Mr. Carr stated that Eagle-Picher has a NASA contract (NAS5-21159) to determine the effects of process variables such as carbonates, loading, etc. on life.

Components should be serialized and material traceability provided so the causes of anomalies can be traced. Retain material test samples in bags in case troubles develop later. Samples should be retained at discrete points in the cell manufacture. A log should be kept on the history of each cell.

The survey also indicated that continuous cleanliness during the processing was essential to reduce the amount of contaminates. All plates, separators and materials should be handled with lint free cotton gloves and sealed in clean room grade plastic bags when not being processed. The process area should be a clean area. The need for gloves is illustrated by the means used to check the plates for pimples, burrs and discontinuities - hands are rubbed across the plates to inspect for projections greater than 0.004 inches.

The previous text on life limiting problems contains recommendations for critical process controls for specific problem areas. To orient those not familiar with the manufacturing processes in making an electrode, the following description is offered.

1) The basic material is either a nickel *screen* (made in France) or a perforated nickel sheet;

2) Selected carbonyl nickel powder is sintered onto the basic material to form a porous sintered *plaque*. It has a porosity of about 80%. The plaque is coined;

3) The plaque is impregnated with nickel nitrate for the positive and cadmium nitrate for the negative electrode. The nitrates are converted into active hydroxides by placing the plaques in KOH and passing a current. The plaque is now called a cell *plate*. It is essential that the weight of active materials on each plate be very close to the mean value to assure capacity matched plates. The amount of active material (and area) of the negative plate is greater than that on the positive plate to prevent the negative plate from obtaining a full charge. Premature charging of the negative plate to full capacity will lead to increased voltages due to the evolution of hydrogen. NASA/Goddard recommends a minimum 1.5:1 negative to positive plate area.

4) The plates are washed, dried and assembled into *cells*, prismatic or cylindrical, and separators are added.

F. PART USAGE CONSTRAINTS

The aerospace Ni-Cd batteries are not normally off-the-shelf items but are procured for a specific mission. Hence, it is not practical to list acceptable and unacceptable batteries. The following general features are desirable for long-life applications:

1) Hermetically sealed to retain electrolyte and prevent contamination of surrounding spacecraft.

2) Voltage monitoring leads for the individual cells to permit monitoring of battery condition for degradation with time:

3) Over capacity batteries for the particular mission is suggested if the weight penalty is permissible. Capacity decreases with age; therefore, an over capacity battery will provide the necessary output longer.

4) The negative plate capacity should exceed the positive plate capacity by 50% to negate the effects of negative plate passivation. Excess negative capacity also decreases overcharge pressure.

5) Pressure relief valves should be used on batteries used on manned missions to preclude crew injury should high battery overpressure occur.

In addition to the battery itself, consideration must be given to maintaining the battery temperature and charge. Temperature controlled environment for the battery will prolong life, a 0°C temperature is recommended. An additional study is recommended to determine the optimum means of controlling battery temperature and means of recharging.

G. REFERENCES AND BIBLIOGRAPHIES

References

1. Lackner, King and Haines: "Battery Design for Aerospace Power Supplies." *Advanced Aerospace Technology Symposium,* Toronto, Canada, March 17, 1969. Canadian Aeronautics and Space Journal, Vol. 15, December 1969, pp. 393-398.

2. J. R. Kent: *Analysis and Evaluation of Spacecraft Battery Life Test Data,* NAD, Crane, Ind., SA-CR-92883, QE/C-67-592.

3. *Transcript of Proceedings: OAO Battery Trouble Shooting,* NASA GSFC, Greenbelt, Md., 31 October 1968, and 17 November 1971.

4. Transcript of Proceedings: *1971 NASA/Goddard-Aerospace Industry Battery Workshop.* Greenbelt, Md., 17 November 1971.

5. D. E. Mains: *Evaluation Program For Secondary Spacecraft Cells - Cycle Life Test Annual Report.* NAD, Crane, Ind., NASA CR-105513, QE/C-69-244.

6. E. C. Bruess: *Evaluation Program for Secondary Spacecraft Cells.* NAD, Crane, Ind., 68/02/02 NASA-CR-94617, QE/C-68-14.

7. S. Gross: *Causes of Failure in Sealed Nickel-Cadmium Batteries.* Boeing Company, Seattle, Washington. Energy Conversion, Volume 11, June 1971, pp. 39-45.

8. D. E. Chusty and J. D. Harkness: *Evaluation Program for Secondary Spacecraft Cells, Seventh Annual Report of Cyclic Life Test.* QE/C 71-1, NAD, Crane, Indiana, January 29, 1971.

9. E. W. Booman, O. L. Linebunk, J. Collum, R. E. Thomas, J. H. Waite: *Study of Space Battery Accelerated Testing Techniques.* NAS5-11594 69/08/00. Batelle Memorial Inst. Columbus, Ohio.

10. T. E. Preusse, *et al:* "Operating Characteristics of Large Size Nickel-Cadmium Cells." *Proceedings of Advances in Battery Technology,* December 4, 1965. Volume 1.

Bibliography

1. DuPlessex: "Nickel-Cadmium and Silver Cadmium Storage Batteries for Space Applications." *Electrochemical Generators for Space Applications.* International Conference, Paris, France. December 4-7, 1967. (21051)

2. W. E. Howden, E. J. McHenry, Y. Okinaka, D. Turner: "Developments Towards an Improved Sealed Ni-Cd Battery." *International Power Source Symposium,* Brighton, Sussex, England. September 20-22 Paper 20.

3. E. Carr: *A 25,000 Cycle Life Battery.* NASA Goddard Space Flight Center Battery Workshop, 1970, pp. 229-241.

4. L. F. Athearn, A. N. Dey: *Long Shelf Life Organic Electrolyte Battery, Quarterly Report.* Mallory and Co., Inc., Burlington, Mass.

5. R. G. Green: *Heat Sterilizable Long-Life Battery Survey.* Martin Marietta, Denver Division, SR-1620-67-6.

6. M. Shoemaker: "Secondary Batteries." *Handbook of Long-Life Space Vehicle Investigations."* Martin Marietta Corporation, M-70-16, December 1970, p. IV-G-1.

7. B. H. Rucker: *Failure Modes and Effects Analysis Task 2, Eagle-Pitcher Battery SAR-8065.* Eagle-Pitcher for Martin Marietta, Denver Division, MCR-69-521 Rev. A, April 1960.

8. *Battery Storage.* Martin Marietta Source Control Drawing PD9400036 (M509).

9. *Analyses and Evaluation of Spacecraft Battery Accelerated Life.* NAD Crane, Indiana. QE/C 70-687, 22 September 1970.

10. *Surveyor Battery Colorimetric and Cycle Life Tests, Final Report.* Hughes Aircraft, Culver City for JPL, 18 March 1969, NASA-CR-100780.

11. S. D. Epstein, J. W. Mauchly, J. Waite: *Computer Methods for the Reduction, Correlation and Analysis of Space Battery Test Data.* Mauchly Systems Inc., Montgomeryville, Pa., NASA CR-98659.

12. J. S. Fordyce, R. B. King, J. McKee, H. J. Schwartz, D. G. Soltis, L. H. Thaller: *Batteries and Fuel Cells.* NASA-LRC, Space Power Systems Advanced Technology Conference. Sp-131, 1966.

13. G. Rampel and I. Weinstock: "Characterization and Optimization of Recombination and Control Electrodes for Spacecraft Nickel Cadmium Cells." *Proceedings of Advances in Battery Technology Symposium,* December 2, 1966, Volume 2.

14. *Extended Abstracts of Battery Division, Volume 10.* October 10-14, 1965.

15. K. E. Preusse, *et al: Parametric Charge Studies for Aerospace Nickel-Cadmium Batteries.* 4th Intersociety Energy Conversion Engineering Conference, Washington, D. C., September 1969.

16. H. Thierfelder and T. Wylie: "Thermal and Mechanical Design of Nickel-Cadmium Aerospace Batteries." *ASME, Space Technology and Heat Transfer Conference,* Los Angeles, June 21-24, 1970, Proceedings Part 1.

17. Gerald Halpert: "A Screening and Selection Method for Nickel-Cadmium Cell Plates." *Electrochemical Society, National Fall Meeting,* Atlantic City, N. J., October 4-9, 1970.

18. Floyd Ford and Thomas Hennigan: "Nickel-Cadmium Batteries for the Orbiting Astronomical Observatory Spacecraft. *24th Power Sources Symposium.* Atlantic City, N. J., May 19-21, 1970.

19. E. Kantner, J. Bene: "Large Capacity Nickel-Cadmium Aerospace Cells of Improved Design." *NAECON 70, IEEE National Aerospace Electronics Conf.,* Dayton, Ohio, May 18-20, 1970.

20. G. Halpert and W. Webster, Jr.: *Secondary Aerospace Batteries and Battery Materials, A Bibliography,* SP 7027. August 1969.

21. DuBuisson, Gizzie and Mueller, Jr.: *A Study of Programs for Evaluation of Component Life*. Martin Marietta Corp., Denver Division, MCR-69-366, August 1969.

22. S. D. Breeskin and A. D. Taylor: "Battery and Third Electrode Performance Characteristics for Varying Charge and Discharge Rates.: and Richard Gaz: "Battery Cell with Third Electrode Performance Test Results.: in *1971 Intersociety Energy Conversion Engineering Conference*, Boston, Mass., August 3-5, 1971.

23. L. Gomberg and H. Thierfelder: "Assuring Reliability in Aerospace Batteries.: *Proceedings Eleventh National Symposium on Reliability and Quality Control*, Miami Beach, Florida, January 1965.

24. R. L. Hadley and A. J. Cototti: "Nickel Cadmium Cells for Optimum Overcharge Capability." *Power Sources Symposium, 22th*, Atlantic City, N. J., May 14-16, 1968.

25. H. Seiger, S. Charlep, S. Lerner: "Electrical Characteristics of a Sealed Ni-Cd Battery Having a Bipolar Construction." *Intersociety Energy Conversion Engineering Conference, 4th Proceedings*, Washington, D. C., September 22-26, 1969.

26. M. Teresa and R. Corbett: "Two Year Flight Performance of the Lockheed Type S1 Ni-Cd Battery." ibid 25.

27. B. Gladstone and B. Newman: "Demand Pulse Charging for Nickel-Cadmium Batteries." ibid item 25.

28. J. McCallum: "Failure Analyses Procedure for Sealed Cells." Annual Power Sources Conference, 23rd, Atlantic City, N. J., May 20-22, 1969.

29. E. Kantner: "Large Capacity Nickel-Cadmium Aerospace Cells of Improved Design." *NAECON 70, IEEE, National Aerospace Electronics Conference Proceedings*, Dayton, Ohio. May 18-20, 1970.

30. B. E. Conway and M. A. Sottar: *Charge, Discharge and Self-discharge Characteristics of Thin Film Nickel Oxide Electrodes*. Performance Forecast of Selected Static Energy Conversion, Denver, 29th Meeting AGARD, Liege, Belgium, June 12-16, 1967.

31. A. H. Reed, P. W. Cover, G. J. McCallum: *Failure Mechanisms and Analysis of Sealed Batteries*. Batelle Memorial Institute, Columbus Laboratories, AFAPL-TR-69-64, September 1969.

32. J. McCollum, R. E. Thomas, and E. A. Roeger, Jr: *Failure Mechanisms and Accelerated Life Tests for Batteries*. Battelle Memorial Institute, Columbus Laboratories, AFAPL-TR-68-83, July 1968.

Industry and Government Survey Contacts

1. Gulton Industries, Metachen, New Jersey, M. Schulman;
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XIII. TRANSDUCERS

by R. W. Burrows and J. S. Janusz, *et al.*

XIII. TRANSDUCERS

A. INTRODUCTION

This study addresses and provides long-life guidelines for temperature, pressure, flow, humidity, oxygen, and carbon dioxide sensors. The latter three types of sensors involve hardware and approaches so diverse that their characterization by a failure mechanism analyses and other very specific analytical approaches is not feasible; therefore, these are discussed from the standpoint of their current development status and their relative desirability for long-life applications.

Transducers may produce their own energy, such as a thermocouple, or they may require a power source, such as a strain gage. Certain transducers require electronic circuitry. In such cases, they may contain either their own integrated electronics, or the electronics may be separately provided - sometimes by the transducer manufacturer, and sometimes by the procuring agency.

The general characteristics of a transducer are:

1) Size, weight, volume;

2) Accuracy;

3) Power requirement;

4) Sensitivity;

5) Stability (sensitivity to drift);

6) Linearity and hysteresis;

7) Response time and recovery time;

8) Susceptibility to environments;

9) Life and reliability, and;

10) Cost.

These are the trade-off parameters which must enter into any selection of a transducer. For example, the most widely used pressure transducer is the potentiometric type. Its very wide usage is due to two factors: It is relatively inexpensive and it has a very low power requirment. On the other hand, it is not the best choice for long-life, high reliability, or severe vibration and shock applications.

Section B presents the long life assurance guidelines. The long-life assurance problems and solutions of each of the six transducer types addressed are presented in separate sections of this chapter as denoted below. Section I presents the results of the industry survey for all six transducer types. Section J presents the references and bibliographies.

Transducer Type	Chapter Section
Temperature	C
Pressure	D
Flow Meters	E
Humidity	F
Carbon Dioxide Sensors	G
Oxygen Sensors	H

B. GUIDELINES FOR LONG-LIFE ASSURANCE

Selected pressure and temperature transducers are capable of life excess of 10 years. Flow meters have less life capability, but meters without moving parts and operating in an environment of low contamination and corrosion are capable of long-life. Humidity, oxygen and carbon dioxide sensors are life limited, requiring periodic maintenance actions such as cleaning, and replacement of cartridges.

A prime problem with transducers is the lack of long-term stability (freedom from drift). The solution to this problem lies in making the total transducer dimensionally stable over long periods of time. Accordingly, it is important to mimimize the use of non-metallic materials and to employ and control processes that yield parts in a stress free condition. Long term stability (and process control) must then be demonstrated through testing of the assembled transducer under appropriate environmental conditions such as temperature cycling.

A well-constructed test program provides the best assurance that a transducer will perform satisfactorily in a long-life application. Unfortunately, the lead time available from selection of a transducer to commitment of the transducer to service may be only a fraction of the time required for a comprehensive test program. Therefore, transducers should be chosen where such data has already been acquired.

Long-life transducer applications fall into two broad classifications: open-loop and closed-loop. Open-loop applications only provide information regarding the performance of a system. Closed-loop applications involve a control function to regulate a system based upon transducer output. Transducer failures in open-loop applications result in uncertainty about the condition of the system, while similar failures in closed-loop applications result in failure of the system. Consequently, closed-loop applications of transducers are more critical from the standpoint of failure effects. They require greater attention to the factors influencing reliability and life.

In general, redundancy techniques do not provide solutions to transducer problems. Active redundancy cannot provide a solution if a known life limiting mechanism exists in the transducer. Multiple potentiometer wipers, exposed to the same wear, may fail within the same time span. Stand-by redundancy is generally not feasible.

In standby redundancy a non-active transducer is protected from the failure producing condition until the first transducer has failed. Applications involving stand-by redundancy are severely restricted by size/weight and the complexity of devices required to switch from the active to the stand-by transducer.

1. <u>General Transducer Guidelines</u>

1) Procurement specifications should be tailored to permit the use of existing designs with proven long-term stability and in current production. Even minor changes to a proven device may invalidate its long-term stability and reliability. Exceptions to this rule should be held to the absolute minimum.

2) A basic cause of problems has been poor technical communication between the manufacturer and the procuring agency with non-technical people in the loop, such as procurement personnel and sales representatives. Timely and efficient communication on technical requirements and problems must be achieved to insure a reliable product.

3) Use hermetically sealed, welded case designs. These are available with almost all types of transducers and are preferred for long-life applications.

4) Designs which eliminate or minimize the use of non-metallic materials are preferred. With many transducers, the creep and/or deterioration of internal non-netallic materials, such as epoxy bonding agents, is a primary life limiting factor. An example of a design approach which eliminates all non-metals is the thin film strain gage pressure transducer.

5) The basic design should minimize stress-induced creep. During manufacturing, annealing processes should be used to further reduce the internal stresses which produce creep and calibration shifts. With many transducers, the creep of metal parts under stress causes calibration shifts.

6) Transducers for long-life applications should be selected, wherever possible, only when long-term, or valid accelerated test data is already available to provide proof of long-term stability.

7) Employ transducers based on simple operating principles and no moving parts since they provide the best promise for long-life applications. Bearings, pivots, gears, and sliders may all be required in a high-performance, high-resolution transducer;

however, the life of such devices will be limited. To achieve long-life, transducer performance may sometimes have to be sacrificed in favor of simplicity.

8) In selecting a specific transducer, the complexity of the associated electronics is frequently the primary threat to long-life. This factor should be strongly weighted.

9) The electronics associated with transducers should be fabricated using screened Hi-Rel parts. If this is not affordable, screening the electronics at the component level using the temperature cycling policy developed in Volume IV, Chapter II, is recommended.

10) Variables introduced in manufacture and assembly of a transducer must be controlled if long-life is to be achieved. Historical data indicate that many transducer failures can be attributed to poor process control, poor workmanship, and inadequate quality control. Poor solder joints and introduction of moisture and other contaminants during manufacture have been problems.

11) The larger devices are usually preferable for long-life and reliability. They are generally more rugged and less sensitive to manufacturing process deviations. Attempts to miniaturize a proven transducer design often compounds workmanship and process control problems.

2. Specific Transducer Guidelines

a. *Temperature Transducers*

1) Platinum wire resistance sensors are preferred for long-life applications. The wire must be of high purity and wound in a manner to minimize mechanical strain.

2) Avoid the use of thermocouples coupled with hi-gain amplifiers because of the problem of thermal instability. Thermocouples are best suited for high temperature measurements.

3) Some thermistors, depending on the specific manufacturer, are subject to drift. Such devices should be stabilized by a 1000-hour burn-in at 150°C.

4) Quartz crystals provide very stable performance, but the complexity of the associated electronics creates a greater chance of random railure.

5) The use of sensistors should be limited to circuit compensation applications because they lack stability as temperature sensors.

6) Life, reliability, and stability should be demonstrated by accelerated temperature cycling with the temperature range and the number of cycles exceeding the projected use conditions.

b. Pressure Transducers

1) The diaphragm should be designed to operate at less than 50% yield to achieve a life of 10^6 cycles.

2) Pressure transducers should be designed with a burst-to-operating pressure margin of 250% minimum to minimize the risk of case rupture.

3) Over-pressure stops should be designed into the transducer to prevent damage to the diaphragm when exposed to pressure surges or transients.

4) The bonding of strain gages is critical because creep of the epoxy will cause drift. Designs which eliminate non-metallic materials are preferred for long-life applications.

5) Potentiometric transducers are not the preferred choice either for long-life applications or for stringent vibration and shock applications because they are susceptible to noise and wear from wiper dither in localized segments of the resistive element.

6) Absolute pressure transducer designs which eliminate feed-through wires into the reference cavity are preferred since they eliminate potential leak paths. Cleaning and sealing of the reference cavity is critical.

7) Pressure transducers should be stabilized by 5 to 20 temperature cycles, depending on the accuracy requirements.

c. Humidity

1) The dew point hygrometer should be limited to applications in which scheduled maintenance is available. The sensing mirror is susceptible to contamination and should be accessible for periodic cleaning.

2) Aluminim oxide hydrometers are not recommended where accuracy and long-term stability are required. Simplicity of the device is offset by its susceptibility to poisoning (and drift) by atmospheric contaminants.

3) Quartz crystal hydrometers with solid sorbents are suited to applications requiring high sensitivity and stability over a wide range of relative humidity.

d. *Flow Meters*

1) Head-type and thermally sensitive meters are sensitive to fluid temperature and require compensation to permit accurate measurement over a wide temperature range.

2) Head-type meters are susceptible to calibration drift resulting from orifice contamination and should be protected with filters in the fluid line.

3) Heat-type meters should be of all welded construction and designed for 250% overpressure to minimize the risk of case rupture.

4) Thermally sensitive meters should be avoided in rapid response applications involving stringent shock and vibration environments. Heater and thermocouple of elements in these meters are fragile and susceptible to damage.

5) Volumetric turbine meters should be restricted to applications in which steady flow is anticipated and fluid temperature is maintained within narrow limits. Thermal expansion of the meter turbine and housing will result in wear, internal leakage, and loss of accuracy.

6) Magnetic flow meters contain no moving parts susceptible to wear and require no obstruction in the fluid stream. They are preferred in applications where maintenance is permitted without disturbing fluid lines.

e. *Oxygen/Carbon Dioxide Sensors*

1) Except for the electrochemical cell, the complexity of instruments available for sensing oxygen or carbon dioxide does not warrant their use in long-term, unattended applications. These instruments include mass spectrometers, infra-red analyzers, ion chambers, spectrophotometers and chromatographs.

2) Electrochemical cell sensors are preferred in applications requiring long-term stability without calibration for 1000 hours.

C. TEMPERATURE TRANSDUCERS

The failure mechanism analysis of temperature transducers presented in Table 1 is discussed in the following paragraphs.

The simplicity of temperature transducers provides a low probability of random failure and, in general, wear-out failure modes are not present. For long-life applications, the measurement stability or freedom from drift is important. The relative importance is a function of the required long-term accuracy. A study by E. E. Swortzlander (Reference 1) directed towards the selection of a transducer for the Skylab telescope mount experiments gives the following data:

Type	Stability
Quartz Crystal	±0.01 deg C per year
Platinum Wire	±0.03 deg C per year
Thermistor	±0.05 deg C per year
Thermo Couple	±1.5 deg C per year
Sensistor	±20 deg C per year

Quartz Crystals

Recently, quartz crystals have been developed with a linear temperature coefficient of oscillation. The crystal is used in an untuned oscillator and the output clipped. The resulting signal can be processed with all digital techniques.

Quartz features an orderly crystalline structure which is noted for its mechanical properties. The sensor is extremely stable and does not require special handling or careful annealing as do many of the other sensors.

A monitor can be designed around this sensor using fairly straightforward digital circuitry. The temperature sensing crystal determines the frequency of an oscillator. The output is mixed with the output of a high stability reference oscillator; the difference frequency is counted for an accurate period of time, and the result displayed. If the reference and nominal sensor frequency are selected to be about 28 MHz, the difference frequency will change by acout 1 KGz/degree C. A 28 MGz system is further attractive because

Table 1 Failure Mechanism Analysis - Temperature Transducers

TYPE	FAILURE MODE	REMARKS	DETECTION METHOD	HOW TO ELIMINATE/ MINIMIZE FAILURE MODES
Quartz Crystals	Associated electronics	Use only for special applications requiring very high stability.	-	Use Hi-rel parts in the electronic circuitry.
Platinum Resistance Thermometer	Open	Breakage due to mishandling damage of very fine wire.	Resistance measurements	Screen by temperature cycling.
		Failure due to thermal incompatibility of bobbin and fine wire.	Resistance measurements.	Include temperature cycling in qualification test.
Thermistors	Drift	Some devices are subject to drift.	Measurements, often high temperature burn-in.	Use hermetic sealing and stabilize by high temperature burn-in.
Thermocouples	Open	Exposure to corrosive mediums.	Test for chemical compatibility.	Select proper metals or provide protective packaging.
		Exposure to high temperatures.	Test for thermal compatibility.	Select proper metals.
		Weld failure.	Pull tests, inspection.	Process control on welding.
	Drift	Impurities cause sensitivity changes.	Chemical analyses.	Specify pure metal.
Sensitors	Drift Open	Wide variation of characteristics among identical units	Functional Test	Don't use as a sensor. Limit use to circuit compensation applications.

of its small size. Repeatable measurements with such a system require a highly stable reference oscillator; this necessitates a temperature stabilized environment for the reference oscillator, which involves a considerable power expenditure.

The complexity of the associated electronic circuitry, as compared with the other types of transducers, results in a much higher chance of random failure; therefore, this approach should be used only for special applications where very high stability is required. The random failure probability for a particular design can be calculated by convential reliability techniques. It can be minimized by the use of circuit redundancy and Hi-Rel electronic parts.

2. Platinum Wire Resistance Sensors

The electrical resistance of pure metals varies with temperature at about 0.2 to 0.5 percent per degree centigrade. The resistance versus temperature characteristic for pure platinum is especially stable and reproducible. The platinum is drawn into a very fine gauge, wrapped around a bobbin support (with a coefficient of thermal expansion similar to that of platinum), and then sheathed in stainless steel, glass, or vitrified ceramic for environmental protection.

The high linearity of the resistance vs temperature characteristics of this sensor is an important feature, but equally important is the extreme stability and reproducibility of this relationship. Commercially available platinum resistance temperature sensors have a repeatability and long-term stability better than ±0.03 degree C at the ice point (0.01 degree C).

This sensor is capable of a life well in excess of 10 years, providing the design is qualified by extensive temperature cycling to verify that the thermal coefficient of expansion differences existing in the packaged unit will not cause loss of the seal, or breakage of the fine platinum wire. A platinum wire sensor can be designed for the same resistance with a few turns of very fine wire or more turns of larger diameter wire. Hence, for long-life and high reliability applications, the largest wire size practical should be used to minimize the chance of failure due to a broken lead either from mishandling or from damage during fabrication. For devices exposed to corrosive media, the sheath should be selected and tested to preclude any long-term corrosion.

3. Thermistors

Technology has not permitted production of consistent and uniform semiconductor resistance sensors until within the last few years. The majority of such devices are produced by sintering various mixtures of the oxides of cobalt, copper, iron, magnesium, manganese, nickel, titanium, uranium, and zinc. Devices of this type are called Thermistors (from thermal resistors) to distinguish them from silicon bulk-effect semiconductor temperature sensors.

Thermistors are exceptionally sensitive, more than an order of magnitude greater than platinum resistance sensors.

The large range of resistance covered by a thermistor makes it nearly impossible to match circuit impedances to them. Also the logarithmic resistance temperature curve shape is a drawback. However, one method of combining both of these negative features in such a way as to achieve nearly linear characteristics over moderately wide temperature ranges is described by Swortzlander (Reference 1).

In the past, thermistors have been relatively unstable devices. Aging of the exposed sintered material caused large resistance shifts, but hermetic sealing of the element has alleviated most of the problem.

JPL has found that high temperature burn-in is necessary to stabilize the devices from most manufacturers. In applications where less than 1% drift is required, their recommended practice is to burn the devices in for 1000 hours at 150°C and then to measure the temperature resistance curve over the desired operating range. The devices are then burned in for 168 hours at 125°C and the temperature vs resistance curve verified. JPL has found this technique to be superior to a temperature cycling test. The stability of thermistors is a strong function of the particular manufacturing processes and the controls on the processes. For example, some manufacturers (such as Yellow Springs Instrument Co.) have developed methods which yield particularly stable devices. When thermistors are procured from such sources, the burn-in, recommended above, may not be necessary.

4. Thermocouples

Common types of thermocouples are chromel/alumel, iron/constantan, copper/constantan, and platinum 10% rhodium/platinum. Thermocouple junctions either may be exposed to the gas or liquid being measured or they may be protected in metal sheaths. For long-life applications, it is always preferable to protect the junction in a hermetically sealed cavity, but this may not be feasible when a high response rate is necessary.

When the thermocouple is selected and applied to avoid corrosion or high temperature degradation, the life is almost unlimited. However, thermocouples are subject to a gradual drift (aging) over a long period of time. Also, slight impurities can cause gross sensitivity changes. These considerations generally limit thermocouple accuracy to ±1.5°F.

Another drawback for a stable long-life is that the low sensitivity of the thermocouple may necessitate high gain amplifiers which are generally subject to thermal instability. For example, a platinum-platinum 10% rhodium thermocouple requires a gain of about 2150 to yield a 0 to 5V for a range of 0 to 70°F. Amplifiers with closed loop gains of much over 100 are generally subject to thermal instability. In addition, thermocouple measurement systems are unwideldy in the sense that a reference junction is required, and the thermocouple wires generally have to be run in the cable harness to the cold junction. It is concluded that thermocouples are preferred for the measurement of high temperatures, but have less utility for measurements over small ranges.

5. <u>Silicon Bulk Resistance Sensor (Sensistors)</u>

The thermal sensitivity of a typical sensor is about 0.7% per degree C, which is roughly twice the sensitivity of platinum wire resistance sensor.

The problem with the silicon bulk resistance temperature sensors is that the units which are commercially available are subject to large variations from one unit to the next. A typical unit has a resistance of 2.7 Kilohm ±5% at 25 degree C, and a resistance of 4.6 Kilohm ±7% at 100 degree C. In addition, the resistance at 25 degree C is subject to variation by as much as ±15% as a result of aging, vibration, shock, etc. Since the sensitivity is about 0.7% per degree C, a temperature measurement could be in error by as much as ±20 degree C anywhere along the curve. A sensor of this type is extremely useful for compensation of transistor parameters which are temperature dependent i.e,, h_{fe} , but is of limited utility as a temperature sensor.

6. <u>Selection Criteria</u>

Important design factors for temperature transducers for long-life are summarized in Table 2. The two most important features, and these are common to almost all types, are that the device should be hermetically sealed. Any device selected should have existing test data which establishes that repeated temperature cycling will not:

Table 2 Selection Criteria - Temperature Transducers

Design Factor	Remarks
A. Quartz Crystal	
Electronics	Since the life and reliability depend upon the associated electronics, these must be fabricated from screened parts, properly packaged, and with thermal control provided to the reference oscillator to avoid temperature drift errors.
B. Platinum Resistance	
Package Design	Insure that the design of the core (usually ceramic) and the wire wrapping are thermally compatible. The device should be hermetically sealed.
Wire Size	Use the largest wire gage feasible to minimize an open failure mode.
C. Thermistor	
Stability	Select the manufacturer whose devices are not subject to drift, or stabilize the devices with high temperature burn-in. Use a hermetically sealed device.
D. Thermocouple	
Package Design	Insure that the package is designed to avoid problems of thermal coefficient of expansion differences. Use hermetically sealed devices with the junctions protected. If response requirements prohibits this, insure the metals selected cannot be corroded by the medium. Thermocouples are low output devices and, hence, the amplifiers must be selected to provide significant gain while maintaining thermal stability.
E. Sensistor	
Stability	Not stable enough to use as a temperature sensor. The main application is in solid-state circuit temperature compensation.

(1) degrade the integrity of the device due to thermal coefficient of expansion differences in the packaging and seal materials, and (2) will not cause a shift in measured output. This latter problem is significant with thermistors, where some manufactured products must be stabilized by a high temperature burn-in, while others are fabricated with special processes yielding a stable device.

D. PRESSURE TRANSDUCERS

Pressure transducers consist of many combinations of sensing element and transduction approaches. The most common combinations are shown in Table 3.

The various sensing devices for pressure transducers illustrated in Table 3 do not comprise a life limiting, fatigue problem when properly designed and applied. Their capability is quoted by nearly all manufacturers as exceeding 10^6 cycles. This is equivalent to 300 cycles per day for 10 years. Even properly designed bellows configurations are capable of long-life. However, they are more apt to acquire a permanent set requiring recalibration. An advantage of the diaphragm-beam type of construction is that the electronics are isolated from possible temperature transients. The material for the diaphragm is usually selected for optimum properties of linearity and hysteresis; but attention must also be given to possible corrosion from the pressure medium, although this is usually not a problem.

In applications where inadvertent overpressures may be applied, diaphragm stops are incorporated to prevent damage to the sensing element.

An absolute pressure transducer is achieved by evacuating the downstream reference cavity. Long-term stability then depends also on maintaining the reference pressure. The preferred design is one which has no electrical feed-through since these are potential leak paths. In addition, the reference cavity should be thoroughly cleaned and outgassed prior to sealing by welding.

The life and reliability is more strongly influenced by the transduction devices. Failure mechanisms are summarized in Table 4 and discussed in the following paragraphs.

Potentiometer

Potentiometer cycle lives quoted by manufacturers are in the order of 10^5 cycles, but these are full-range cycles and are not necessarily indicative of their true life capability. For example, the total travel of a potentiometer is typically about 0.2 inches. A typical potentiometer is wrapped using 0.4 mill wire. During storage and handling, the transducer is exposed to daily pressure and temperature changes, transportation on rough riding carts and trucks, etc. This causes wiper dither and very localized wear on

Table 3 Summary of Common Pressure Transducer Design Approaches

Sensing Devices → ← Transduction Devices

Sensing Devices

Flat Diaphragm

Corrugated Diaphragm

Diaphragm-Beam

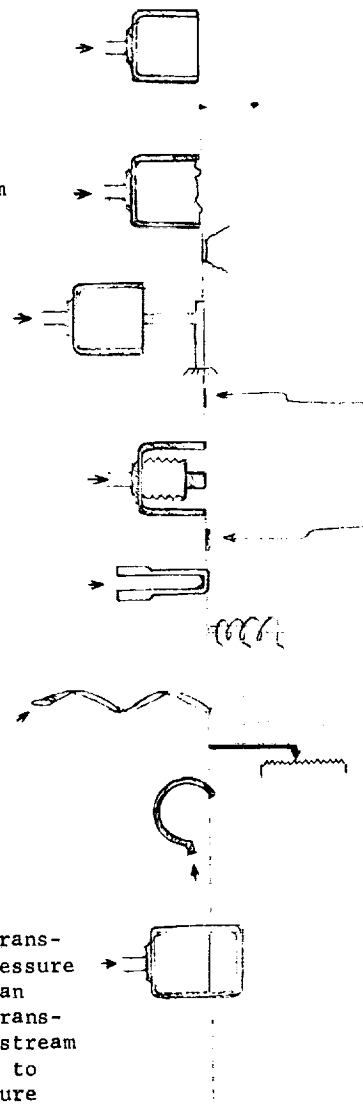
Bellows

Straight Tube

Twisted Bourdon Tube

Circular Bourdon Tube

Absolute Pressure Transducer. Any gage pressure transducer becomes an absolute pressure transducer when the downstream cavity is evacuated to the reference pressure (usually 0)



Transduction Devices

Unbonded Strain Gage

Bonded Strain Gage

Thin Film Strain Gage

Silicon Strain Gage

Variable Reluctance and Differential Transformers

Potentiometer

Table 4 Failure Mechanism Analysis - Pressure Transducers

PART	FAILURE MODE	REMARKS	DETECTION METHOD	HOW TO ELIMINATE/ MINIMIZE FAILURE MODES
Sensing Element and Transducer Body	Leak		Leak Test	Use welded construction. Avoid feed throughs into the reference cavity.
	Burst		Proof Pressure Test	Design for 250% over-pressure.
	Calibration Shift	Can be caused by: Inadvertent overpressure	Calibration tests	Design in overpressure stops
		Diaphragm fatigue	Life tests	Design to operate at less than 50% yield.
		Creep	Calibration tests	Anneal to remove residual stresses in metal.
Tranducing Element (except potentiometric)	Calibration Shift	Usually caused by the instability of the bonding cements	Temperature and pressure cycling	Eliminate all non-metallic materials such as epoxy bonding cements.
	Open	Usually caused by interconnection failures of fine wires	Functional test	Good process control on wire bonding. Welded connections are preferred.
	Short	Very uncommon. Devices are not highly susceptible to particle contamination	Functional test	Clean rooms not deemed essential. Assembly can performed on a clean bench.
Potentiometric	Noisy, erratic	Prove to wear from wiper dither of small segment of resistance winding	Functional test	Avoid using for long-life applications.

one small segment of the winding. This may also loosen contaminating materials and particles. Most failures are due to this cause rather than full range cycling. Obviously, this phenomena is most marked with the low pressure transducers.

Other disadvantages of the potentiometric types are that they are more subject to overpressure damage. Their damage threshold is in the order of 200% of full range, as compared with 500 to 2000% for other types. This is because of their long travel (0.2 inch) as compared with the other types. Also, their tolerance for vibration and pyrotechnic shock is significantly less than for the other types, and is in the order of 15 to 20g rms (vibration) and 3000 to 5000g (shock) for the other types.

Despite these disadvantages, potentiometric types probably represent 90% of all transducers used. Their widespread usage is due to two factors. They are very economical, but their primary advantage is their low power requirement, about 1 ma at 5 volts. However, current trends in instrumentation are tending to favor the adoption of the other, longer life types. For example, three years ago a nominal power requirement for a strain gage type was about 200 ma, but currently this has been reduced to about 10 ma. Instrumentation systems which switch the devices on and off, as required, are becoming more feasible. It therefore seems reasonable that for long-life applications, the potentiometric apporach should be discouraged in favor of the other, longer life types. It should be mentioned, however, that some potentiometers have provided relatively long service. Genesco Technology supplied one to Honeywell that lasted 8 to 10 years, and Servonics has had potentiometers in Saturn since 1966 that still function. In the final analysis, however, potentiometers cannot be deemed the best choice for long-life applications.

With respect to the incidence of so-called "random" failure in potentiometric types, a paper (Reference 2) by John A. Wickham reports on the analyses of 265 transducer failures on a launch vehicle 1959 to 1967. His data, (Figure 1), shows the failure rate of potentiometric pressure transducers was greater than for variable reluctance types and much greater than resistance thermometers.

2. Unbonded Strain Gage

Unbonded strain gages utilize wire wrapped over ceramic terminals and held in place by a non-metallic bonding agent. One set of terminals is fixed and the opposing set connected by welded beam structure to pressure diaphragm. Significant zero shift and

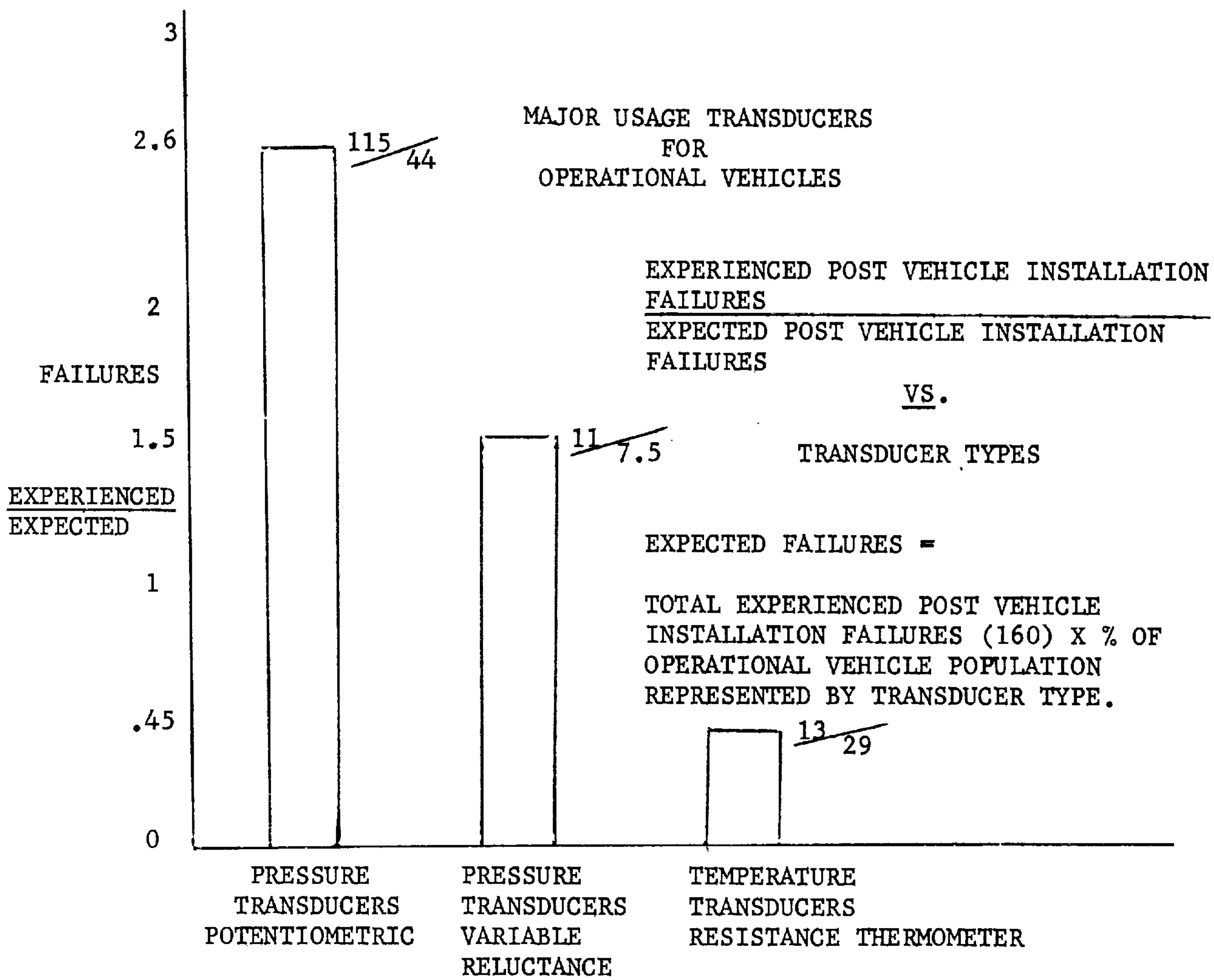


Fig. 1 Transducer Failures on a Launch Vehicle Program

sensitivity changes occur during initial temperature and pressure cycling due to stress relief of the welds and minor creep of the bonding agent. Once past this initial relaxation period, however, zero shift and sensitivity changes will remain within acceptable tolerance for design environments.

The life of this device is essentially unlimited providing the non-metallic bonding agent is stable and properly applied.

3. Bonded Strain Gage

The weakest link, for long-life, is the non-metallic bond between the strain gage and the sensing diaphragm. Every pressure cycle "stretches" this bond and the bond may creep under load. This deterioration is aggravated in high temperature applications. A very stable bonding agent, properly applied, is probably adequate for 10 years, but this failure mechanism can be bypassed by using the thin film approach where the strain gage is vacuum deposited on a ceramic substrate and no organic material is involved.

4. Thin Film Strain Gage

The motivation in the development of the thin film strain gage was the elimination of the organic bonding agent between the strain gage and the sensing elements. Pressure transducers using thin film strain gages can be entirely constructed of metals and ceramics, a desirable feature for very long-life. In this process, an insulating ceramic film is first vacuum deposited on the metal pressure diaphragm. Four strain gages are then vacuum deposited on the ceramic substrate. Attaching leads are then vacuum deposited between the four gages forming a bridge. In the last step, the connecting wires are welded to the device, using the same technology as employed for wire bonding of integrated circuits, except the quality of each bond is subjected to much closer inspection than would be tendered on an integrated circuit.

5. Silicon Strain Gage

The advantages of the silicon strain gage are the high gage factors and the very small size. Gage factors from 50 to 200 are typical, either positive or negative. For long-life applications, epoxy bonding of the silicon to a metal sensing diaphragm is not recommended. The problem is that silicon has a low coefficient of expansion and with high temperature curing of the epoxy, a residual compression stress exists in the silicon. Creeping of the bond may subsequently occur, resulting in a severe measurement drift. Kulite

Semiconductor Products is currently developing bonding techniques which eliminate the use of organic materials. For example, the silicon could be bonded to the metal with glass frit. This problem is avoided if the entire pressure diaphragm is a silicon disc, but problems of chemical incompatibility may exist with some pressure media such as hydrazine.

When the silicon itself is used as the pressure diaphragm, this bonding problem is bypassed and, in addition, this approach permits the fabrication of extremely small devices. A pressure sensing disc 1/4 inch in diameter is considered large by one manufacturer who has produced devices 0.030 inches in diameter. Their high gage factor adapts them to measurements of dynamic phenomena, where the pressure fluctuations are in the region of 100,000 Hz and higher. In most applications of these devices, long-life has not been a consideration; but there is no inherent reason why long-life devices could not be fabricated. Molecular diffusion of the dopant into the surrounding pure silicon could conceivably create a long-term drift problem, but this is not believed to be a problem at temperatures under 200°C.

6. Variable Reluctance and Differential Transformers

This class of transduction devices operate by using mechanical movement obtained from a Bourdon, bellows, diaphragm, or U-tube in such a manner as to vary the inductance or reluctance of an electromagnetic circuit.

The typical variable reluctance device consists of a magnetic diaphragm mounted equally distant from two E-shaped magnetic cores with one isolated winding on each core. These inductors are placed in an ac bridge and, as pressure moves the diaphragm, a voltage unbalance is produced by the change in air gaps. This gives a relatively high level of electrical output for a very small mechanical travel. Full scale travel of the diaphragm is usually in the order of 0.003 inches. Material selection is somewhat restricted in that the diaphragm is required to be magnetic. The typical device also requires wire penetrations of the pressure cavity because the E cores and windings are located in the cavity.

Differential transformers commonly consist of a separate magnetic core within the field of a transformer that has one primary and two secondary windings mounted on a single spool. The magnetic core is moved by a sensing element, thus changing the magnetic coupling between the primary and each secondary. Alternating current is supplied to the primary, and voltages are induced in each secondary

proportional to the displacement of the core. Differential transformers require mechanical connection of the magnetic core to the sensing element, but can have complete metallic isolation of the pressure medium with all welded seals. The transformer windings are then mounted on a non-magnetic spool that is part of the pressure cavity with the magnetic core inside the spool.

Electrical opens, caused by inter-connection failures of fine wires, are the most common failure modes for these devices. The E core in the variable reluctance transformer and the magnetic core in the differential transformer require protective covering if used for sensing pressure of corrosive fluids or gases. A non-conductive material is used to seal the E core from the pressure cavity. Gold plating is used to protect the magnetic core from the corrosive effect. The coil of the differential transformer is located outside the pressure medium and therefore does not require special consideration.

Both transduction devices utilize non-metallic insulation and bonding materials which may introduce di-electric breakdown and loss of bond strength as a result of aging. Aging effects of high temperature environments can also reduce permeability of the magnetic materials, thus changing the sensitivity of the device with time.

Genesco Technology predicts an operating life for variable reluctance devices of two years or 2×10^7 cycles and a five year storage life. Differential transformers are rated by Gulton Industries, Inc., Servonics Division, at five years operating life, unlimited cyclic life, and five years storage life. Several other manufacturers indicated that the design of these two types of devices is well suited for long-life usage. Present life limits are governed by existing applications data and should be extended in the foreseeable future.

Table 5 summarizes the important features to consider when selecting a pressure transducer for a long-life application.

Table 5 Design Factors for Long-Life Assurance of Pressure Transducers

DESIGN FACTOR	REMARKS
Packaging	Select welded, hermetically sealed designs.
Wire Insulation	Use Kapton insulated wire. Teflon insulated wire subject to shorting from cold flow.
Wire Size	Avoid use of very small gage wire. Use largest practicable size. Gages smaller than No. 40 to 50 are fragile and more susceptible to breakage.
Material Compatibility	Insure the pressure cavity material will not be corroded by the fluid or gas to be measured.
Strength of the Pressure Cavity	Design with a burst-to-operating pressure margin of at least 250%.
Non-metallic Materials	Select transducers which do not contain, or have minimized the use of non-metallic materials.
Size	Avoid use of small size designs. These foster increased workmanship and process control problems.
Wear-out Life	Avoid use of potentiometric transducers for long-life applications.
Vibration	Avoid use of potentiometric transducers in stringent vibration and shock environments ($>$ 20 g's rms vibration, 200 g's shock).
Temperature Transients	If the pressure cavity is exposed to severe temperature transients, isolate the transduction device by using a diaphragm-beam configuration.
Reference Cavity Design	With absolute pressure transducers, select a design which has no feed-through wires or other potential leak paths.
Over-pressurization	If inadvertent overpressures may be encountered, select a design with diaphragm restraints to prevent damage.
Stability (drift)	Select a developed and proven design for which the manufacturer can provide comprehensive long-term test data on stability (drift), and evidence of a good field failure history.

E. FLOW METERS

Flow meters are categorized by their application, either gas or liquid. Gas flow instruments include head-type meters and thermally sensitive meters. Liquid flow instruments provide a wider range of choice and include volumetric turbine meters, direct-mass meters, and magnetic flow meters. Failure mechanisms are summarized in Table 6 and discussed in the following parahgraphs.

1. Head-Type Meters

Head-type meters indicate volumetric flow by measuring the fluid pressure drop across an orifice (in the case of thick plate or knife-edge orifices) or the upstream pressure (in the case of nozzles) given a prior calibration of the meter using the appropriate gas. The principal reason for distinguishing between these three types is the repeatability of the discharge coefficient under different flow conditions.

Head-type meters will undergo performance degradation due to discharge coefficient drift as a result of ambient temperature variation. This condition will correct itself when ambient temperature control is regained.

Head-type meters will not react to mechanical shock or vibration environments. However, damage to the upstream and downstream pressure sensing connectors from these mechanical loads will cause erroneous outputs which are irreversible.

Head-type meters are most sensitive to fibrous material which will establish a trap for all contaminants resulting in calibration changes. This is an irreversible change without maintenance (cleaning), but the meter will be re-useable after maintenance. Redundant orifice meters will reduce risk.

Head-type meters will survive indefinitely, with tolerance for some performance degradation. Extensive non-aerospace applications involving gas flow with high contamination levels have noted as much as five years' service between maintenance periods.

Table 6 Failure Mechanism Analysis--Flow Meters

TYPE	FAILURE MODE	REMARKS	DETECTION METHOD	HOW TO ELIMINATE/ MINIMIZE FAILURE MODES
1. Head-type	Calibration shift	Can be caused by: Ambient temperature change	Calibration tests	Provide temperature compensation
		Particulate contamination	Calibration tests	Use redundant orifice meters
	Pressure sensor leak		Leak test	Use welded construction
	Pressure sensor burst		Proof pressure test	Design for 250% over pressure
2. Thermally Sensitive	Broken element	Can be caused by: Particulate contamination	Functional test	Contamination control of fluid
		Vibration/shock	Functional test	Avoid using for long-life applications
	Open	Usually caused by interconnection failures of fine wires	Functional test	Good process control on wire bonding
	Calibration shift	Caused by ambient temperature change	Calibration test	Provide temperature compensation

Table 6 (cont)

	TYPE	FAILURE MODE	REMARKS	DETECTION METHOD	HOW TO ELIMINATE/ MINIMIZE FAILURE MODES
3.	Volumetric Turbines	Bearing seizure	Can be caused by: particulate contamination	Functional test	Contamination control of fluid
			Over-speed	Functional test	Avoid using in systems subject to pulsing or surging
		Bearing/seal leakage	Can be caused by: vibration/shock	Functional test	Provide shock isolation
			Ambient temperature change	Functional test	Provide specific thermal control
		Calibration shift	Can be caused by: particulate contamination	Calibration test	Contamination control of fluid
			Ambient temperature change	Calibration test	Provide specific thermal control
4.	Direct Mass Meters	Bearing/seal leakage	Can be caused by: vibration/shock	Functional test	Provide shock isolation
			Ambient temperature change	Functional test	Provide specific thermal control
		Calibration shift	Can be caused by: particulate contamination	Calibration test	Contamination control of fluid
			Ambient temperature change	Calibration test	Provide specific thermal control
		Open	Drive motor electrical connections due to vibration/shock	Functional test	Good process control on wire bonding

Table 6 (concl)

TYPE	FAILURE MODE	REMARKS	DETECTION METHOD	HOW TO ELIMINATE/ MINIMIZE FAILURE MODES
5. Magnetic Meters	Calibration Shift	Caused by fluid density change due to ambient temperature change	Calibration test	Provide specific thermal control
	Open	Electrical connections due to vibration/shock	Functional test	Good process control on wire bonding

2. Thermally Sensitive Meters

Thermally sensitive meters operate on one of two principles:

1) Measurement of heat loss from a continuously heated probe enables inference of the fluid velocity.

2) Measurement of the temperature difference between two points upstream and downstream of a heated section in the flow stream permits inference of the mass flow.

The heated probe meter is designed primarily to provide rapid response and so is virtually a point sensor which is insensitive to variations in temperature across the flow stream. Power requirements are small but the sensor is easily damaged.

The heater and thermocouple elements in the heated flow stream meter, variation (2), are similarly fragile, particularly the thermocouples, and power requirements are significantly greater. Various specialized configurations have been devised to divert the flow into smaller conduits to reduce power requirements, but always at the expense of external electronics simplicity.

New variations of the heated probe meter involve orificing the flow to minimize stratification errors; however, risk of damage from particulate contamination is increased.

Thermally sensitive meters must be equipped, as part of their construction, with ambient temperature compensation sensors. In spite of such compensation, ambient variations will induce thermal stratification in the flow path that cause meter inaccuracy.

To achieve fast response, thermally sensitive meters contain inherently fragile elements. These elements including electrical connections, that are susceptible to physical breakage often because of particles in the flow path. Also suspect are the sealed electrical connections from the flow path to external controls.

Heated probe meters will survive one to two years, given clean gas to monitor and periodic maintenance/checkout of the relatively complex, external, electronics. Non-maintainable aerospace applications to date have been limited to short duration usage. Their dependence on external power implies an electrical interface which in itself could limit life.

3. Volumetric Turbine Meters

Volumetric turbine meters depend on fluid velocity causing turbine rotation which is counted digitally, integrated by external electronics and converted to volumetric flow. The nature of turbine/housing relationships is such as to require zero leakage past circumferenctial clearances as an objective. Degradation of this clearance will result from contamination abrasion and thermal expansion of the turbine into the housing. Resulting drag on the turbine will cause erroneous output and, if the interference condition is removed, a larger than desired clearance will cause permanent calibration change. Rotor bearings operate in the fluid medium; and while this frequently provides cooling, it also exposes bearings to fluid chemical and particle contamination.

Volumetric turbine meters must be protected against excursions in the fluid ambient temperature by specific thermal control. Increased turbine/flow conduit clearance permits internal leakage which results in performance failure (inaccuracy); reduced turbine/flow conduit clearance will result in either marginal turbine interference (inaccurate output) or total failure due to siezing.

Volumetric turbine meters require isolation of the flow conduit from the mechanical excitation of shock and vibration to avoid damage to bearings and sealed couplings of the digital counter. Bearings and radial clearances will first be degraded (calibration shift) and ultimately may sieze due to particulate contamination. The turbine conduit clearance particularly will not be renovated by maintenance because once contaminants have intruded and the turbine rotation has been impeded, the conduit surface would be damaged.

Given some contamination control of the fluid and adequate ambient thermal control, turbine and Impeller meters will have a life expectancy of one to two years depending on the duty cycle. This limitation reflects the joint effects of bearing degradation (random contamination), rotating seal wear, and random turbine overspeed effects.

4. Direct Mass Meters

Direct mass meters impart directive momentum to the fluid and then use a turbine to remove the momentum. Measurement of the mass effect of the momentum results in a measure of flow. A variation of this technique uses external power to maintain a constant impeller torque and uses impeller speed as a measure of the mass effect. Both types use external motors, have fluid-to-ambient rotating seals, bearings in the flow stream as well as statically sealed couplings from the flow stream to flow readout.

Direct mass meters (momentum transfer) suffer from increased external leakage due to ambient temperature variations when the flow path is coupled through rotating seals to external drives or elastic links.

Direct mass meters will be most sensitive to shock and vibration at the flow path coupling seals, at shaft bearings, and at drive motor electrical connections. Bearings are susceptible to particulate contamination with degraded rotor response occurring first and total failure following eventually.

5. Magnetic Flow Meters

Magnetic flow meter operation is based on the principle that electrical currents are induced in a conductor which is put in motion through a magnetic field. Fluid volume flow generates a voltage that is proportional to the volume flow rate. These meters have the advantage of using no moving parts and no obstructions in the fluid stream.

The response of magnetic meters is influenced by temperature changes in the fluid stream, principally through fluid density effects. Unpredictable density variation results in meter inaccuracy. Magnetic meters are insensitive to particulate contamination and will survive the life of the controlling external electronics, which should not be limited by any wear-out effects. Table 7 summarizes the important features to consider when selecting a flow meter for a long-life application.

Under conditions of low Reynold's Number sub-sonic flow, (low volumetric flow, small flow paths or large orifice to pipe diameter ratio) heated probes are preferred if the gas is clean relative to particulate and chemical contaminants. If one of these contamination conditions is a specific threat or could develop after long use, then thick plate orifices are preferred for long-life applications.

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Table 7 Design Factors for Long-Life Assurance of Flow Meters

DESIGN FACTOR	REMARKS
1. Gas Flow Applications	
a. Flow Conditions	
Low Reynolds Number	Heated Probe meters provide best accuracy with no particulate contamination. Use Thick Plate Orifice meter if contamination is anticipated.
High Reynolds Number	Thick Plate Orifice meters are preferred.
Critical Flow	Thick Plate Orifice meters are preferred.
b. Particulate Contamination	Use Head-type meters if contamination could develop.
c. Vibration	Avoid use of fragile thermally sensitive meters in stringent vibration and shock environments.
d. Temperature Transients	Provide ambient temperature compensation to reduce calibration drift.
e. Wear-Out Life	Avoid use of fragile thermally sensitive meters for long-life applications.
2. Liquid Flow Applications	
a. Flow Conditions	
Single Phase Liquid	Volumetric Turbine meters provide best accuracy for steady flows. Use Thick Plate Orifice meters for pulsing or surging flows.
Two Phase Liquid-Gas	Direct Mass meters are required with Angular Momentum type preferred.
Two Phase Liquid-Solid	Gyroscopic Momentum meters are preferred.
b. Particulate Contamination	Magnetic Meters are the only type insensitive to contamination.
c. Vibration	Select Magnetic Meters for stringent vibration and shock environments.
d. Temperature Transients	Meter inaccuracy occurs in all types with ambient temperature extremes.
e. Wear-Out Life	Use High-Rel electronics with Magnetic Meters for long-life applications.

Thick plate orifice meters are preferred in High Reynolds Number ($>10^5$) non-critical flow applications. The nominal repeatability of a thick plate orifice flow coefficient, assuming no mechanical or chemical degradation of the orifice, is only 2 per cent. If greater accuracy repeatability is required, knife-edge orifice meters are recommended.

Thick plate orifice meters are preferred under critical flow (Mach $\overset{\sim}{\geq}$ 1) conditions. Although flow nozzles can be used in critical flow applications, their increased construction complexity and the relationship of this construction (shape and dimension) to the metering function make them less desirable for long-life usage. The nozzle flow coefficient is more reactive to low level particulate or chemical contamination.

Turbine meters are recommended under single-phase liquid flow conditions for applications in which metering accuracy must be better than 0.5 per cent. However, if flow conditions are such that pulsing or surging is possible and if the flow Reynolds Number will be large ($>10^5$), then a thick plate orifice must be used. Pulsed flow through an orifice will cause only a transient failure, but in a turbine this flow environment can cause overspeeding and siezing. If the liquid flow is characterized by large particulate contaminants, then accuracy control must be compromised since magnetic meters are the only recommended solution for long-life operation. Corrosive liquid flow will drastically degrade performance of any of the candidates except thick plate orifice or magnetic meters.

Applications involving two-phase liquid gas or cryogenic flow conditions require direct mass meters. The angular momentum type is the simplest and is recommended. In this type, angular momentum is imparted to the fluid by a driven impeller and the momentum is removed by a turbine. Accuracy is good, less than 0.25 per cent error; but this meter presents high risk for long-life applications. The constant speed impeller motor is coupled into the flow stream through a rotating seal as is the elastic restraint on the turbine. Contamination in the fluid will threaten the accuracy of the turbine response or possibly cause failure of the response by curcumferential siezing. Surging or pulsed flow also can cause turbine overspeed and, in turn, siezing of the turbine. If any of these abnormal influences are possible, then the only recommended meter is the gyroscopic momentum meter. This meter has an acceptable accuracy but is non-linear in comparison with angular momentum meters. It is both mechanically (highly-stressed rotating seals) and electrically complicated and does not promise long-life capability without frequent maintenance and recalibration.

The only recommended meter in two-phase, liquid-solid (slurry) flow applications is the gyroscopic momentum meter. Any direct mass or columetric meter employing a turbine is inappropriate because of the solid phase which acts like particulate contaminants. The orifice meters likewise will not be adaptable for the same reason. The disadvantages of this meter cited above must be accepted and a significant risk to long-life operation expected.

F. HUMIDITY TRANSDUCERS

Humidity transducers represent a greater problem for long-life applications than do temperature or pressure transducers.

Three types of sensors are discussed, the Dew Point Hygrometer, the aluminum oxide hygrometer, and the quartz crystal hygrometer.

1. Dew Point Hygrometer

This device is being used in the Skylab Orbital Workshop. The unit contains a mirror surface which is thermally bonded to a thermo-electric cooling module. The module, when excited with direct current of the proper polarity, causes cooling of the mirror. When the mirror reaches the dew point temperature, condensation begins on the mirror surface. The presence of condensation on the mirror changes the visable light reflection characteristics of the mirror. The mirror is illuminated by an incandescent source such that a change in reflectivity is detected by the direct and bias photocells. The photocells are connected as legs of a bridge circuit. The bridge output is connected to an operational amplifier whose output is used to control the current supplied to the thermoelectric cooling unit. The system stabilizes and controls about a particular dew layer thickness. A thermistor is used to measure the stabilized mirror temperature. The thermistor is used as the active leg of a bridge which provides the input to an operational amplifier. The amplifier output is conditioned so that a dc voltage proportional to mirror temperature is obtained.

When the stainless steel mirror first collects a layer of contamination, an adjustment of the optical bridge will null out the contamination effect. However, when the mirror collects a sufficient layer of contamination to cause an impairment in accuracy, the mirror must be cleaned.

This sensor is manufactured by Cambridge Systems Inc. who supplied the following life information. The Skylab version is a third generation device. An earlier version was used on Apollo. The incandescent light source used on earlier versions has been replaced by a light emitting diode, eliminating the long-life problem of filament wear-out. The light sensors are cadmium selenide photoresistors hermetically sealed in TO-18 cans and protected from overload damage by Zener diodes. The mirror is gold-plated silver. The

silver provides high thermal conductivity and the gold is protection against tarnishing and corrosion. The thermistor in the mirror, per NBS data, will remain stable within 0.2°C for a period of five years. The weak-link from the standpoint of very long-life is the thermoelectric cooler. The repeated thermal cycling will eventually break down the bismuth telluride joints. Cambridge Systems Inc. does not have formal life test data but reports that a similar dew point sensor in an industrial application functioned continuously for 2½ years before failure of the thermoelectric cooler from the repeated thermal cycles.

One maintenance consideration exists. When the sensing mirror becomes contaminated, it must be cleaned. This is facilitated by removal of a single nut to remove the sensor which can then be cleaned with a small swab such as a Q-tip.

The following information was obtained by the Skylab user of the device, McDonnell-Douglas Corp. A sensor is currently under mission-simulation tests and has been exposed to 28 days operating, 34 days storage, and 56 days operating, under a simulated Skylab environment of 5 psi atmosphere of O_2, N_2 and CO_2. The unit performed very satisfactorily except for some evidence of possible measurement degradation near the end of the 56th day exposure. This problem is currently being evaluated. The mirror is cleaned prior to final installation in the inlet duct to the molecular sieves. In this location, the sensor is not accessible for cleaning. Future Skylab experience with the device will provide further information about the device and whether easier access for maintenance is desirable.

2. <u>Aluminum Oxide Hygrometer</u>

These sensors are small, simple, and economical; but there is considerable question about their long-term stability. The sensor consists of a plate of anodized aluminum to which electrical connections are made to the base metal and the aluminum oxide layer.

During the oxidation or anodizing process on aluminum, certain other chemicals are entrapped in the aluminum oxide layer making the layer hydroscopic. The absorbed moisture changes the impedance of the sensor and yields a measurement of humidity. However, there is a considerable question about their accuracy and their stability with time. Reports on long-term stability from various manufacturers are conflicting. The device is apparently susceptible to atmospheric contaminants which poison the aluminum oxide layer causing calibration shifts. The sensor is also temperature sensitive.

This device is attractive from the standpoint of simplicity and cost, but the issue of accuracy and long-term stability should be examined in detail prior to selecting the device for long-life applications.

3. Quartz Crystal Hygrometer

Piezolectric quartz crystals are used extensively for controlling frequency in communications equipment, and are widely used as selective filters in electric networks. Special quartz crystals are available that can control frequencies to one part in 10^9. It is estimated that commercially available 15 MHz crystals will have a sensitivity of about 2600 Hz per microgram. Hence, if the crystal is coated with a hydroscopic layer which changes mass in relation to humidity, the frequency of oscillation will depend upon the humidity.

The frequency of such a coated crystal depends on the mass of the coating, which in turn depends on the mass of vapor absorbed by the coating.

Since sorption isotherms of many materials are often known, the performance of a sorption detector is readily predicted. Liquid-coated crystals make rapid and linear response detectors, but solid sorbents are outstanding for high sensitivity at low concentrations.

The use of deliquescent salts for coating crystals results in interesting water detectors. The simplest coating procedure is vacuum evaporation through a mask so that only the electrode receives the film. The detectors are extremely rapid at normal conditions, but at low temperatures and low humidities the response can be very slow due to the lack of driving force, slower diffusion, and the existence of the solid state. For example, a lithium chloride coated crystal whose coating thickness was only 500 Å had a time constant of 18 minutes at -26°F and less than 1 second at room temperature providing the direction was dry to wet. In the reverse direction the hydrates must be contended with, and about 6 minutes are required to dry down.

Water detectors can be made from a wide variety of materials, and each has its own range of usefulness. Gjessing *et al.* (Reference 3) developed a radio-sonde humidity element consisting of an SiO_2 evaporated film on a crystal. They report no hysteresis between 15 and 95 per cent RH. King (Reference 4) has observed similar

results with silica gel coated crystals; however, hysteresis did show up below 5 per cent RH. A selective water detector based on a hygroscopic polymer coated crystal has now been commerically available since 1964. The device is currently marketed by the Instrument Products Division of E. I. duPont de Nemours & Co. Inc. The unique qualities of the instrument are parts-per-million detection in 30 seconds, high selectivity, and long life.

G. CARBON DIOXIDE SENSORS

There are several techniques available for monitoring CO_2 levels in a breathable atmosphere. The ion chamber and electrochemical cell type instruments have been qualified and used in space flight environments. Design and development have been pursued on the infrared and the mass spectrometer type analyzers for space flight application.

1. Ion Chamber Sensors

The Skylab CO_2 Detector System is a radiation counter type instrument which was originally designed for and used on the Gemini spacecraft.

This type of instrument uses filters, radioactive material and electronics to compare the ion content of two streams of gas. One gas stream is filtered to remove water and carbon dioxide. The other gas stream is filtered to remove only water. Both gas streams are then directed into ion chambers where the radioactive material ionizes the gases. Each ionized gas stream is then passed between two plates which are excited by dc voltage. The ionized gases provide ion currents between the plates. Since the ion currents of the two gas streams are of opposite polarity, the currents are summed together to provide a difference in reading which indicates the amount of CO_2 in the gas sample. The difference in ion currents is changed to a dc voltage and is then amplified to a 0 to 5V dc output for readout.

The filter units used with this instrument are life limited and a filter change is required for every 10 days of operation in the Skylab environment.

2. Electrochemical Cell Sensors

A portable CO_2 Sensor Assembly using an electrochemical sensor was developed by Beckman and qualified for the Apollo Program. The sensor accurately measures CO_2 concentration and provides an alarm if the CO_2 level rises to hazardous levels.

The sensor consists of a glass pH bulb and a silver chloride reference electrode. The sensor surface is coated with a thin film of buffer (bicarbonate) which is held in place with a very thin semipermeable membrane of silicone rubber.

This system measures a range of 0.1 to 30mm Hg CO_2 and is capable of maintaining long-term stability without calibration for 1000 hours. With periodic calibration, the sensor life is expected to be in excess of one year of operation. The associated electronics are relatively simple and provide high reliability. The device is small, lightweight and requires less than one watt to operate. A comparable CO_2 sensor using the electrochemical cell principle was also used in the life support system of Project Mercury.

3. Non-Dispersive Infrared Analyzer

The infrared analyzer provides a continuous determination of the concentration of CO_2 in a gaseous or liquid stream. The unit uses two light beams which pass through a reference cell and a sample cell. The reference cell contains a known concentration of CO_2 and the sample cell contains the gas or liquid requiring analysis. Infrared radiation which passes through the sample cell is absorbed by the CO_2 component only in the wavelength regions where the CO_2 has infrared absorption bands. The amount of absorbed radiation is proportional to the CO_2 concentration in the sample.

This type of analyzer was designed by Beckman and considerable development effort was accomplished to produce a prototype instrument for monitoring concentrations of carbon monoxide and methane in the MOL Program.

4. Mass Spectrometer

The Mass Spectrometer is an instrument which is capable of simultaneously measuring more than one component (O_2, CO_2, N_2, etc) in a multicomponent gas. The instrument can be made selective for a single component analysis and a continuous measurement can be obtained. The output can be displayed, recorded, or used to provide control for gas management systems. Examples of these instruments include the following:

1) The Quadrapole Mass Spectrometer is one form of the mass spectrometer which has been recommended for development of a long-life instrument capable of monitoring and controlling the partial pressure of oxygen and trace contaminants in a spacecraft atmosphere. This type of instrument uses an electron multiplier detector. Secondary emission is produced at the alkaline earth surface of a collector plate. The electrons are collected on a series of dynodes which are similar to a photo-multiplier. The output is amplified and can be displayed, recorded or used for control functions.

2) The Beckman Metabolic Gas Analyzer is a quadrapole mass spectrometer with a digital computer which was designed and developed for NASA for determining physiological performance of individuals. The instrument is capable of measuring the partial pressures of oxygen, carbon dioxide and other gases, as well as providing data on oxygen consumption, carbon dioxide production flow rate and respiratory quotient.

3) The Perkin-Elmer Metabolic Gas Analyzer (P/N 10M13040-3) which also utilizes a mass spectrometer for analysis was developed for NASA. This instrument is presently being qualified for use in medical experiments in the Skylab Program.

   In their present forms, these instruments are adapted for use by an individual while performing some physical task. The instruments are large, heavy, complex, and require on the order of 125 watts of input power.

Adaptation of one of this type of instrument to the function of atmosphere monitoring and control would require studies and further development to achieve smaller size, reduced weight, and reduced power consumption. The complexity of these instruments would require evaluation to determine if long-life can be achieved, particularly with unattended operation.

H. OXYGEN SENSORS

Numerous instruments are available for measurement and control of oxygen levels in a breathable atmosphere. Some of these instruments are simple devices using electrochemical cells. Others are complex instruments such as mass spectrometers, paramagnetic analyzers, spectrophotometers, and gas chromatographs.

1. Electrochemical Cell O_2 Sensors

There are presently two spaceflight qualified electrochemical type O_2 sensors available for measuring the partial pressure of oxygen in spacecraft atmosphere. These are the polarographic type and the galvanic type.

These electrochemical sensors are life limited due to depletion of some portion of the sensor during exposure to oxygen. Sensor lifetimes of up to one year of service can be expected.

a. The MOL Oxygen Analyzer - It utilizes an electrochemcial cell and an amplifier to measure the oxygen content in a breathable atmosphere.

The sensor is a polarographic type electrochemical cell which consists of a gold cathode and a silver anode which are covered by an electrolyte (potassium chloride) contained by a FEP teflon membrane. A calibrated bias voltage is applied to the sensor. When the membrane surface is exposed to the atmosphere, the oxygen diffuses through the membrane and causes a change in the output voltage in proportion to the oxygen partial pressure.

In the process of chemical reaction between O_2 and KCL, the electrolyte becomes depleted at a rate proportional to O_2 concentration. Service life of the sensor is in excess of 1.25×10^6 mm Hg-hrs (PO_2 measured in millimeters of mercury times the number of hours exposed.) At a PO_2 level of 200 mm Hg, this equates to over 6250 hours.

The orignial version of this type of system was qualified and flown on the BIOS Program. Additional development and testing was done for the MOL, Gemini and Skylab Programs.

The system using the electrochemical type sensor is very reliable, requires very low power, is small in size, lightweight and relatively uncomplicated to use. It provides excellent long term stability and does not present a safety hazard. The unit can withstand exposure to vacuum without adverse effects.

Present problems with the sensors are associated with membrane material. Suitable quality control and screening tests have been instituted to minimize or eliminate these problems. However, the search for improved membrane material is an area presently being investigated.

b. The Skylab O_2 Detector System - It utilizes an electrochemical cell and a converter/amplifier to measure the oxygen content in the breathable atmosphere.

The sensor is a galvanic type electrochemical cell which consists of two electrodes covered with an electrolyte (potassium hydroxide) held in place by a teflon membrane.

During exposure of the membrane surface to an atmosphere containing oxygen, the oxygen diffuses through the membrane at a rate dependent upon the quantity of oxygen available. The oxygen combining with the KOH results in production of a voltage which is proportional to the amount of oxygen in the sample.

In the process of producing the output voltage, the copper electrode is consumed at a rate dependent upon the total oxygen diffused through the membrane. This process causes the sensor to be a life limited item which must be replaced for each manned phase of the Skylab mission.

This sensor unit, as with dry cell batteries, has a shelf life limit, even when not exposed to oxygen. This limit results from corrosion of the copper electrode by the KOH electrolyte by the KOH electrolyte. At present, the shelf life is limited to 12 months.

This system, using the electrochemical type sensor is considered very reliable, requires very low power, is small and lightweight and is simple to operate. The unit can withstand exposure to vacuum without adverse effects. The present problems with the sensors are associated with the membrane material. Quality control and screening tests for completed sensors have been instituted to eliminate acceptance of units with marginal membranes and other defects.

To provide longer life sensors, a manufacturer of the galvanic units has recommended an increase in sensor size to allow an increase in the size of the copper electrode. It was also pointed out that improved membrane material is being sought.

c. The Spacecraft Cabin O_2 Analyzer - It was developed for and used successfully on the Apollo Program. The analyzer utilizes a polarographic sensor similar to that described for the MOL Oxygen Analyzer. The electronics package provides meter type readout and a capability for measuring oxygen partial pressure in ranges of 0-300 and 0-800 mmHg. The system can be used for either cabin O_2 or spacesuit O_2 analysis. Sensor limitations, problems and improvement requirements are the same as those mentioned for the MOL oxygen analyzer.

d. The Aircraft Oxygen Monitoring System - It was developed for integration into combat aircraft. The system is another in the family of instruments which utilize an electrochemical sensor for detection of oxygen level in a breathable atmosphere. Although not proven for space flight, the instrument is cited here as another example of the extensive application of the electrochemical sensor in monitoring a critical parameter in a simple and reliable manner.

2. Mass Spectrometer

The Mass Spectrometer is an instrument which is capable of simultaneously measuring more than one component (O_2, CO_2, N_2, etc) in a multicomponent gas. The instrument can be made selective for a single component analysis and a continuous measurement can be obtained. The output can be displayed, recorded or used to provide control for gas management systems. A discussion of these instruments in contained in Section G. Their capabilities and limitations are equally applicable to O_2 and CO_2.

3. Paramagnetic Analyzer

The Paramagnetic Amplifier has potential for applications in long-term measurement and control of oxygen. Although present analyzers of this type have been used for more than 15 years, development work is required to provide a unit capable of withstanding spacecraft environments.

The Paramagnetic Analyzer provides an indirect determination of oxygen by measuring the cooling effect of oxygen on a heated wire within a magnetic field. Direct measurement of oxygen can also be provided by measuring the amount of displacement that a test body makes in a magnetic field as a result of the presence of varying amounts of oxygen.

4. Other O_2 Sensors

Other instruments which can be used to detect and measure oxygen levels are the Spectrophotometer and the Gas Chromatograph. However, in view of the complexity of these instruments and the lack of recommendations for their use, further inquiry was not pursued.

I. INDUSTRY SURVEY

An industry survey (Table 8) of both transducer manufacturers and users was conducted to determine the reasons for the success of specific transducers used in past programs. Information from fourteen companies was compiled; seven manufacturers and seven users. Four transducer types were selected for this survey: temperature, pressure, flow meter, and accelerometer. The information accumulated fell into two categories; technical and nontechnical.

The technical information involved materials, design, fabrication, aging, testing, and inspection. The manufacturers acclaimed that the materials used in the design represented no compromise with cost. The materials were selected upon the basis of low wear and long life. Metallic materials were highly preferred to non-metallic materials. Some manufacturers reprocessed raw materials to enhance the purity and quality. The purity of platinum wire used in temperature transducers, for example, has a dramatic effect upon long-term accuracy and the usefulness of the device.

The design of the transducer was oriented to provide a device which would maintain long-term accuracy in the anticipated environment. Replacement and maintenance was minimized by reducing the number of moving parts and reducing the normal wear of moving parts. For example, a turbineless flow meter, which used the principle of relating the frequency of swirling gas to units of flow, could assure the customer years of trouble free performance.

Most of the manufacturers used very sophisticated fabrication methods. Temperature sensitive pressure strain gages were meticulously wound in a manner as to provide minimal mechanical stresses in the wire element. Element wires were welded to terminals to provide maximum strength and insure low contact resistance. Some devices, such as the pressure transducer, were hermetically sealed in a double metal case by the use of electron beam welding. Welding is believed to assure the device of a more lasting seal and a surer encasement for the active element. The inner case was fabricated from platinum to minimize the presence of impurities in the immediate vicinity of the active element and thereby minimize self-contamination at high temperature.

All transducers were aged at high temperature and most were cycled between low and high temperature limits to improve stability and long-term accuracy.

The devices were tested following the aging procedure. The transducers were calibrated at room temperature. The calibration was checked at the working temperature extremes. Temperature compensation was made to the transducer until its calibration was stable at temperature extremes. Auxiliary tests, such as vibration and shock, were generally conducted on request only. The transducer piece parts, assemblies and completed device, received 100% physical and functional inspection. The transducer user selected the transducer on the basis of the past performance and good reputation.

The users almost always selected the transducer manufacturer on the basis of his demonstrated ability to produce stable, reliable, devices. The users were not concerned with the initial cost of the transducer as much as they were with the long-term accuracy (stability) and reliability. Low maintenance costs were felt to justify a higher initial cost.

There was non-technical information obtained from this survey which should be mentioned. Of the seven manufacturers surveyed, six stressed the need for better communication between manufacturer and the customer. The aspects of communication emphasized the most were the following:

1) Manufacturers frequently lacked information concerning the customer's application of the transducer which limited the manufacturer's usefulness in determining the correct transducer from his product line and making technical suggestions.

2) Manufacturers sometimes misinterpreted the requirements of the customer because messages were relayed through a purchasing agent instead of directly from the customer's engineer.

3) Manufacturers felt that changes in the customer's transducer requirements are frequently received too late to avoid manufacturing and scheduling impact.

Table 8 Survey of Users and Manufacturers

PART DESCRIPTION AND USING PROGRAM	USERS OPINION OF WHY PART WAS SUCCESSFUL	MANUFACTURER'S OPINION OF WHY PART WAS SUCCESSFUL	RECOMMENDED GUIDELINES, IMPROVEMENTS, PROCUREMENT, SPEC REQUIREMENTS, ETC.	RELATIVE IMPACT ON YIELD & COST OF INCORPORATING RECOMMENDATIONS
Flow meter, electromagnetic and turbine types. Used in Executive Aircraft Co. and Navy Target Drones.	1. Accurate and had low maintenance. 2. Local representative of transducer manufacture was technically knowledgeable and provided good maintenance service. 3. Complete technical documentation to the customer minimized transducer installation problems.	1. Designed for low maintenance. 2. Insisted on reliability and long-term accuracy. 3. Used quality components. 4. Sustained a reputation for quality by releasing a new device for sale only after extensive testing.	1. Salesmen should be trained engineers. 2. Salesmen must be honest, give all technical facts, leaving nothing unsaid for the customer to find out for himself. 3. The salesman must know his product line thoroughly.	1. Impact is minimized if the salesman is knowledgeable and honest with the customer.

Table 8 (cont)

PART DESCRIPTION AND USING PROGRAM	USER'S OPINION OF WHY PART WAS SUCCESSFUL	MANUFACTURER'S OPINION OF WHY PART WAS SUCCESSFUL	RECOMMENDED GUIDELINES, IMPROVEMENTS, PROCUREMENT, SPEC REQUIREMENTS, ETC.	RELATIVE IMPACT ON YIELD & COST OF INCORPORATING RECOMMENDATIONS
Flow meter, magnetic (liquid) and swirl (gas). Used in JPL Programs.	1. The capability of the company to design, test and deliver a suitable device was demonstrated in the past and was selected for that reason. 2. The magnetic and swirl type flow meters had long life because of being manufactured with high quality materials and having no moving parts.	1. The transducer had no moving parts. 2. The electronic system was burned-in to identify defective components. 3. The transducer was adapted to a wide range of applications.	1. The vendor and customer require close communication to create a cost effective program. Technical meetings should be held between user and manufacturer at onset of program, but additional meetings only as necessary thereafter. 2. Determine the company's rejection rate and the rework methodology; this has bearing on ultimate quality. 3. Use company customer references to determine previous buyer's opinion of manufacture. 4. Manufacturer should make available his schematics and drawing to the customer.	1. Efficient information transfer would reduce cost.

Table 8 (cont)

PART DESCRIPTION AND USING PROGRAM	USERS OPINION OF WHY PART WAS SUCCESSFUL	MANUFACTURER'S OPINION OF WHY PART WAS SUCCESSFUL	RECOMMENDED GUIDELINES, IMPROVEMENTS, PROCUREMENT, SPEC REQUIREMENTS, ETC.	RELATIVE IMPACT ON YIELD & COST OF INCORPORATING RECOMMENDATIONS
Temperature transducer, thermocouple type. Used in Titan vehicles and small aircraft engines.	1. Company had excellent communication with the customer which resulted in securing the best device for the required application. 2. The type of thermocouple transducer which was mounted in a sealed metal tube had long life.	1. Complete exclusion of atmospheric moisture and moisture in magnesium oxide that surrounds the thermocouple wires which were sealed in a stainless steel protective tube. Sealed-in moisture causes corrosion which shortens the life of the thermocouple element.	1. Maintain good and honest communication between the vendor and customer engineers. 2. Improve communications between the customer purchasing agent and the customer engineer. The purchasing agent receives information from the vendor, but there is excessive delay before the customer engineer is relayed the same information and takes required action. Such a delay should minimized.	1. The delay in communication in the customer's house delays important technical decisions and increases costs. Good communication saves money.

Table 8 (cont)

PART DESCRIPTION AND USING PROGRAM	USERS OPINION OF WHY PART WAS SUCCESSFUL	MANUFACTURER'S OPINION OF WHY PART WAS SUCCESSFUL	RECOMMENDED GUIDELINES, IMPROVEMENTS, PROCUREMENT, SPEC REQUIREMENTS, ETC.	RELATIVE IMPACT ON YIELD & COST OF INCORPORATING RECOMMENDATIONS
Accelerometer, spring mass and force balance types. Used in Nike Missile, Autopilot for Century Series and Lance Program.	1. The requirements of the buyer closely met the specifications of the vendor's catalog, so a proven device could be utilized without any redesign. 2. Simplicity of the device. 3. The history of the company indicated a producer of quality devices.	1. Simplicity of device was important for long-term accuracy. 2. Minimal moving parts. Example: Variable reluctance sensor will outlive the wiper/potentiometer sensor. 3. The test most emphasized was that for accuracy and linearity following acceleration cycling. 4. High quality materials. 5. Thorough inspection during assembly. 6. Cleanliness in assembly of device.	1. Better device installation instructions should be supplied to the customer. Devices returned to the vendor show unreasonable installation abuse which results in devices losing their calibration. Beefed up devices are available by vendor, but include a weight penalty. 2. Vendor should know more about the customer's device application. This would save the vendor and customer money. The customer frequently over-specifies his requirements. 3. The vendor should be accurate in stating delivery dates.	1. It would be cost-effective for the customer to understand the physical limitations of the device during installation to avoid damaging it or causing its accuracy to be reduced. Better installation instructions should be a low cost item. 2. Better communication between vendor and customer would reduce costs and provide a more realistic program schedule.

Table 8 (cont)

PART DESCRIPTION AND USING PROGRAM	USERS OPINION OF WHY PART WAS SUCCESSFUL	MANUFACTURER'S OPINION OF WHY PART WAS SUCCESSFUL	RECOMMENDED GUIDELINES, IMPROVEMENTS, PROCUREMENT, SPEC REQUIREMENTS, ETC.	RELATIVE IMPACT ON YIELD & COST OF INCORPORATING RECOMMENDATIONS
Temperature Transducer, platinum wire type. Used in Skylab, Mariner, Nimbus, Viking Orbitor, Titan Centaur.	1. The transducer supplier was carefully selected on the basis of his demonstrated good performance. 2. The transducer had previously demonstrated long time stability.	1. High purity of platinum temperature element was used to provide a known and low drift temperature/resistance characteristic. 2. The wire element was wound by a method which provided minimal mechanical strain of the element. 3. Reliable bonding to active element was used. 4. Provided thermocycling to establish reliability and promote stability of the transducer. 5. Calibration is degradated by contaminants and is greatly worsened at high temperatures. Precautions were made to seal the transducer from offending environment. The active element was encased in platinum to limit self-contamination at high temperatures.	1. The buyer should confide to the vendor the detailed usage of the transducer so that the vendor may understand the requested transducer specifications and have an opportunity to make suggestions to the buyer. 2. The buyer should not specify a transducer in such a way that it would require a completely new design. The vendor feels that a minor modification to his catalogued devices is all that is generally required. 3. Early planning and dialog between the buyer and the vendor should be encouraged to produce a cost effective program.	1. The cost and impact would be reduced by establishing early information exchange between the buyer and vendor at the onset of a new program.

Table 8 (cont)

PART DESCRIPTION AND USING PROGRAM	USERS OPINION OF WHY PART WAS SUCCESSFUL	MANUFACTURER'S OPINION OF WHY PART WAS SUCCESSFUL	RECOMMENDED GUIDELINES IMPROVEMENTS, PROCUREMENT, SPEC REQUIREMENTS, ETC.	RELATIVE IMPACT ON YIELD & COST OF INCORPORATING RECOMMENDATIONS
Pressure transducer, strain gage type. Used in Titan III and Hawk Missile.	1. The manufacturer was selected because of his past competance. Subsequently, excellent communication prevailed, the manufacturer gave great attention to small design details, and never invoked the excuse of "proprietory information".	1. Thorough testing to assure accuracy of device. 2. Designed device rugged enough to take much abuse. 3. Manufactures own piece parts wherever possible. 4. 100% inspection of piece parts and stages of device assembly. 5. Rigid temperature compensation adjustment procedure to assure long-term accuracy.	1. Stress importance of right device for application. Example: A semiconductor type has high sensitivity, but could never be as accurate as the wire strain gage type. 2. For best vendor/customer relationship, the two should interface early in the program. 3. The vendor/customer relationship is better when the vendor engineer is permitted to communicate directly to the customer engineer and not indirectly through the purchaser or salesman. 4. Prevent misinterpretation of specifications. Misinterpretation is frequently the case when communication between the vendor and the customer is minimal.	1. The vendor/customer schedule impact is reduced by early scheduling and frequent honest communication between vendor/customer engineers and vendor salesmen to customer purchaser.

Table 8 (concl)

PART DESCRIPTION AND USING PROGRAM	USERS OPINION OF WHY PART WAS SUCCESSFUL	MANUFACTURER'S OPINION OF WHY PART WAS SUCCESSFUL	RECOMMENDED GUIDELINES, IMPROVEMENTS, PROCUREMENT, SPEC REQUIREMENTS, ETC.	RELATIVE IMPACT OF YIELD & COST OF INCORPORATING RECOMMENDATIONS
Pressure transducer strain, gage type. Used on Apollo.	1. Thin film strain gage was a superior design to the other types because of its long-term stability. 2. Only few calibration checks were required.	1. Only highest quality materials were used without compromise with cost. 2. State-of-the-art manufacturing methods; electron beam welding was used exclusively during assembling. 3. Temperature cycling at maximum operating temperature limits until device is stabilized. 4. Aging at maximum temperature and operating power for 168 hours week improved stability. 5. The active element was completely isolated from external contaminants by improved design.	1. The customer should keep the vendor updated with any required changes to prevent last minute modifications which would impact inventory and manufacturing schedules. 2. The customer does not rely enough on the experience and know-how of the vendor. The customer should provide all necessary requirements for the device and rely upon the good judgment of the vendor to produce a satisfactory device.	1. Uncoordinated modification requests result in unnecessary delay in delivery of device.

J. REFERENCES AND BIBLIOGRAPHY

References

1. E. E. Swartzlander: "Comparison of Temperature Sensors for Apollo Experiment Instrumentation." *Proceedings of the 24th Annual ISA Conference*, Ball Brothers Research Corp, 1969.

2. J. A. Wickham: *Transducer Failure Effects and Vehicle Operations.* General Dynamics Convair, ISA Index 0103322.

3. D. T. Gjessing, C. Holm, and T. Lanes: *Electronic Letters 3.* No. 4, 1967.

4. W. H. King: *Analytical Chemistry.* Volume 36, Page 1735, 1965.

Bibliographies

J. C. DuBuisson, et al: *Handbook of Long-Life Space Vehicle Investigation.* M-70-16. Martin Marietta Corporation, Denver, Colorado, December 1970.

Thermistors, Descriptions, Applications, Stability Data. Fenwal Electronics, Inc.

"Fundamentals of Aerospace Instrumentation - Vol 1." *Tutorial Proceedings of the Fourteenth International ISA,* June 3-5, 1968.

K. S. Lion: *Instrumentation in Scientific Research.*

"Advances in Instrumentation." *Preceedings of the 24th Annual ISA Conference,* Vol 24, Parts 1, 2, 3, & 4. Houston, October 27-30, 1969.

22nd Annual ISA Conference Proceedings, Volume 22, Parts I and II. ISA, Chicago, 1967.

Proceedings of Instrument Society of America, Twelfth National ISA Aerospace Instrumentation Symposium. Philadelphia, Penn, May 1966.

O'Higgins: *Basic Instrumentation, Industrial Measurement.* McGraw-Hill, 1966.

International Congress on Instrumentation in Aerospace Simulation Faciliites. IEEE Publication, July 1971.

John W. Dean: "Temperature Effects on Pressure Transducers." *ISA Transactions,* July 1966.

Contamination Control Handbook. Sandia Laboratories, NASA CR-61 264, February 1969.

George Horn: *Pressure Measuring Devices for Extremely Low Gas Densities.* Ilikon Paper, APS 64H.

R. N. Blazey, J. R. Schneider: *High Response Laser Flow Meter, Final Report.* AFRPL-TR-69-202, May 1969.

M. A. Perry, Editor: "Flight Test Instrumentation." *AGARD Conference Proceeding No. 32,* Montreal, Canada, 30 May - 1 June 1967.

"A Cross Section of Opinion. . .Instrumentation Forecasts for the 70's. *Instrumentation Technology,* January 1970, p. 41.

Cole & Reger: "Humidity Calibration Techniques." *Instruments and Control Systems,* January 1970. p. 77.

Glawe, Krause & Dudzinski: "Temperature, Pressure and Flow Direction Probe." *Instruments and Control Systems,* January 1970, p. 109.

J. M. Benson: "Instrumentation Technology." *Journal of ISA,* July 1971, p. 39.

M. P. Wilson, Jr.: "Survey of Mass Flowmeters" *Instrumentation Technology* (Journal of ISA), September 1971, p. 63.

K. C. Cotton, Editor: *Flow Measurement Symposium.* ASME Symposium, Pittsburgh, Penn, September 26-28, 1966.

Harry Norton: *Transducer Handbook.* Prentice-Hall, 1970.

Herman K. Neubert: *Instrument Transducers; An Introduction to Their Performance and Design.* Oxford, England, Claredon Press. 1963.

J. J. Martin: *The Role of Instrumentation and Testing in Space Mission Reliability.* Engineering Laboratories and Services, Missile and Space Systems Division, Douglas Aircraft Company, ISA Index 0103325.

Earnest H. Ehling: *Range Instrumentation.* Prentice Hall Inc 1967.

Introduction to Transducers for Instrumentation. Statham Instruments, Inc 1966.

Peter R. Perion: *Thin-Film Strain Gage Transducers.* Statham Instruments, Inc 1965.

Reliability Stress and Failure Rate Data for Electronic Equipment. MIL-HDBK-217B.

Failure Analysis Reports on the Titan III Space Vehicle. Martin Marietta Corporation.

John Wickham: *Transducer Failure Effects and Vehicle Operations.* General Dynamics Convair, ISA Index 0103322.

Vibrating Diaphragm Pressure Transducer. NASA SP-5020, August 1966.

W. Crawford Dunlap: *New Component Developments for Space Technology.* 19th Congress of the International Astronautical Federation, N. Y., October 1968.

I.S.A. Transducer Compendium. Instrument Society of American, 1963 and 1969.

Dr. M. A. Griffin, and J. Rorez: *Philosophy and Practices of Reliability as Applied in the Design of the Saturn Instrumentation System.* University of Alabama and NASA/MSFC, ISA Index 0103321.